POWER

Volume 63 New York, May 18, 1926 *No.* 20

Progress in Power Plant Design

IN 1891 George H. Barrus, M. E., whose practice among the industries of the East was sufficiently broad to make its summary representative of the conditions of the time, published a book on "Boiler Tests."

In the case of the water-tube boilers less than 14 pounds of coal was burned on an average per square foot of grate surface and less than 0 .3 of a pound per square foot of evaporating surface per hour and 10.58 pounds of water was evaporated per pound of combustible.

We are not doing very much better in efficiency today, but Hell Gate burns five times as much coal and evaporates about five times as much water per square foot of heating surface per hour. Surface subjected to radiant heat evaporates 40 pounds or more per square foot and the present tendency is to put more and more of the heating surface where it can see the fire.

In 1900 Mr. Barrus published another book, "Engine Tests." The average steam rate of the condensing engines reported was 21, and of the non-condensing engines 32.7 pounds per horsepower-hour, which is about 28 and 44 pounds per kilowatt-hour respectively.

Turbines are now sold under guarantee to produce a kilowatt-hour on less than 10 pounds of steam.

True, to make the higher-pressure superheated steam requires more heat per pound, but not in anything like this proportion. Reduced to a heat basis, 28 pounds of steam at 100 lb. gage worked to a vacuum of 26 inches would be equivalent to over 30,000 B.t.u. per kilowatt-hour. Modern stations are running on less than half of this.

The average steam pressure in the 60 engine tests that Mr. Barrus records was 101 and the highest 167 pounds. There were a few cases of the use of superheat, the highest 82, the average 41 degrees.

Today pressures above 350 pounds and temperatures over 750 degrees are part of the day's work and a successful installation is running with an initial pressure of 1,200 pounds.

How far should we have got on this road without the specialization and training that the demands of the central station have developed and the realization of which has been so effectively furthered through the committees and resources of the N.E.L.A.?

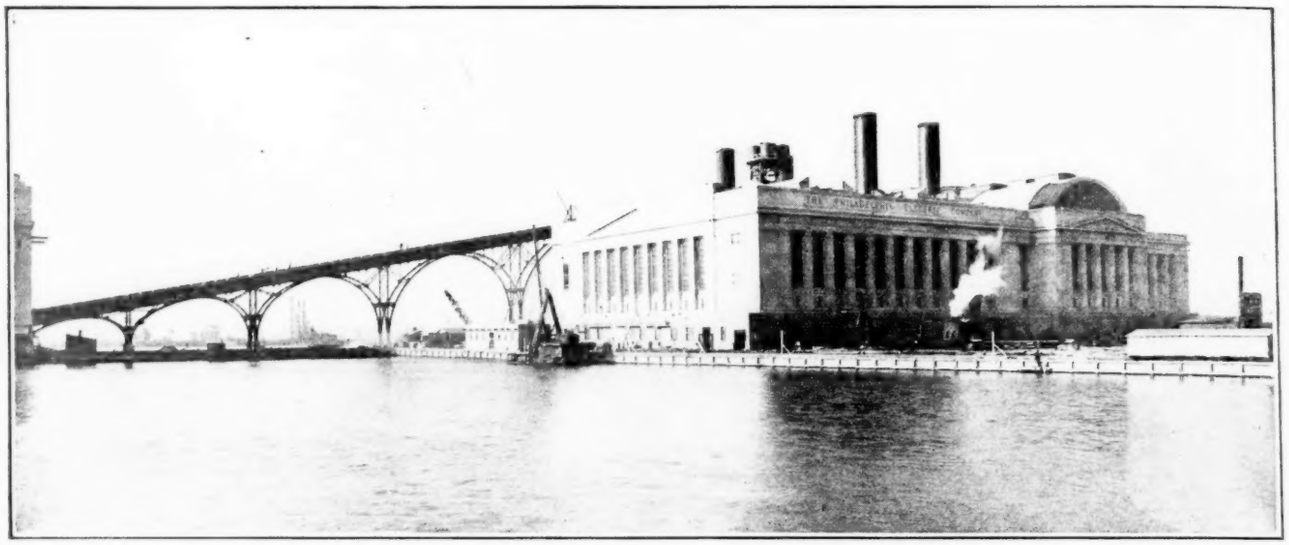


Fig. 1—View of Richmond station from the Delaware River

Philadelphia Electric Company's Richmond Station

Station Designed for 600,000 Kw. in Three Independent Sections of 200,000 Kw. Each —100,000 Kw. Installed—Boilers of 15,697 Sq.ft. Used with Economizers and Air Heaters in Same Settings — Steam Conditions 375 Lb. Gage, 675 Deg. F. — Three-Stage Bleeding from Main Units and Motor-Driven Auxiliaries, Starting at Full Voltage

IN 1922, seeing that the capacity of its present generating stations would soon be reached and that the growth of load would demand a large increase in generating capacity, the Philadelphia Electric Company laid plans for a new station to supplement its other generating stations, which include Schuylkill, Chester and Delaware, with a total capacity of 425,000 kw. The site of the new station, which comprises a 64-acre tract of undeveloped land along the Delaware River about three miles above the Delaware station, was selected as being admirably adapted for the development of a large steam-power station. It is in the midst of a rapidly growing industrial section, the territory to the north and south being already well occupied. The river at the point adjacent to the plant affords a good supply of circulating water as well as good facilities for receiving large shipments of coal.

The station is designed for an ultimate capacity of not less than 600,000 kw. to be built in three sections. In laying out the plant in three independent units it appeared that not only would the reliability be increased, but the light and ventilation would be materially improved. Construction work was begun on the station April 25, 1924, and on Nov. 16, 1925, 18 months later, the first unit was put into service. The design of the station, which was made by the company's engineers in co-operation with John C. Windrim, architect for the building, embodies a number of features of particular interest and indicates an advanced step in power-plant design.

In the present section there are two 50,000-kw. units served by twelve inclined-tube boilers of 15,697 sq.ft. of effective heating surface each, fired by underfeed stokers. This section of the plant is designed to accommodate two more units of similar capacity which will be added as the demand grows.

Previous to laying plans for the station or making any decisions as to the steam pressures and temperatures to be carried, types of equipment, etc., investigations were made of a number of large central stations both in this country and abroad. With the information and data obtained, a complete study was made of several arrangements embodying the following conditions: Steam pressure, from 250 to 550; all using about the same total temperature; average yearly vacuum; temperatures of feed water; various methods of heating feed water; various types of auxiliary drive; different types of boilers with and without economizers and preheaters and with various combinations of both. With the investigations completed, the thermal efficiency and capital cost of a number of combinations of pressures and heat-balance arrangements were worked out and applied to load conditions on the system. From this survey it appeared that a steam pressure of 375 lb. at the turbine throttle in combination with a heat-balance system employing three-stage bleeding and large air heaters, would give the lowest cost per kilowatt-hour. As it was not considered practical at the time the station was designed to use the high air temperatures that would result from the use of large air-heater surfaces

with underfeed stokers, the combination offering the next lowest cost per kilowatt-hour was selected. This called for 375 lb. throttle pressure with three-stage bleeding, using boilers with approximately 150 per cent air-heater surface and 50 per cent economizer surface. Extensive tests previously conducted on a similar combination at the Chester Station had demonstrated the practicability of this particular combination.

The selection of the inclined water-tube boiler was influenced to a large extent by the satisfactory results obtained with this type of boiler over a period of years in the other stations. Also, it appeared to work in best with the general plant layout and the desirability of combining the economizer and air heater within the boiler setting, in place of putting them in independent settings.

In considering the matter of steam pressure ror the plant, investigation showed that there was little to choose between 400 and 600 lb. The initial cost of the station would be much higher for 600 lb. pressure, and this additional cost would not be compensated by the increased efficiency obtained. Also, the fact that the load factor may become poorer as the station grows older, favored 400 lb. pressure. It was also appreciated that as 400 lb. represents the terminal pressure for the high-pressure cycle (1,200 lb.), it would work in well should it be found desirable to go to higher pressures in the future additions to the station. The steam conditions finally decided upon were therefore 400 lb. at the boiler, or approximately 375 lb. at the turbine throttle with an average total temperature of 675 deg. F.

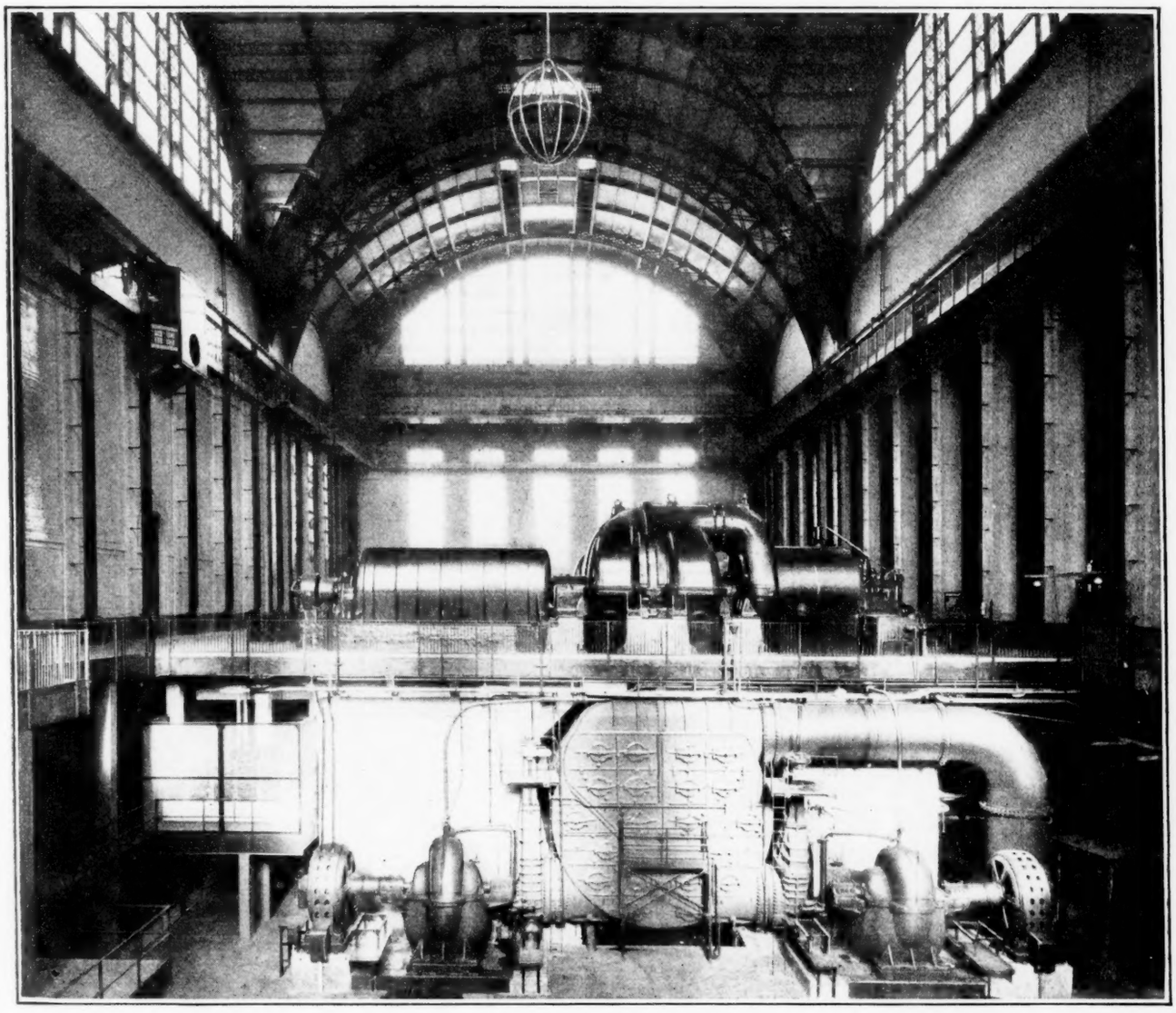


Fig. 2—General view of turbine hall, showing 50,000-kw. tandem-compound unit

As a large number of large central stations designed previous to the Richmond plant had adopted pulverized fuel as the method of firing the boilers, the reason for selecting underfeed stokers for Richmond are of interest.

As the coals commercially available in the Philadelphia territory are of comparatively high grade, which can be burned with practically the same efficiency on stokers as in pulverized form, the ability to utilize a low-grade fuel was not a factor, owing to the higher transportation charges for an equivalent B.t.u. value produced. Also, from the investigation made it was found that with the use of preheated air and adding

together the operating and maintenance charges, the cost of current at the bus would be at least as low with stokers as with pulverized fuel. There was also the question of eliminating the ash dust from the products of combustion emitted from the stacks if pulverized coal were used, for at the time the station was designed no satisfactory solution of this problem had been found. In addition to the foregoing, the large preparation plant that would be required, was considered by the engineers an added complication to the operation of the plant as well as materially increasing the capital cost.

It should be understood, however, that neither the question of higher pressures and temperatures nor that of burning pulverized coal is considered closed. As the successive sections are added, such engineering improvements will be incorporated as experience shows to be warranted or advancement in the art dictates.

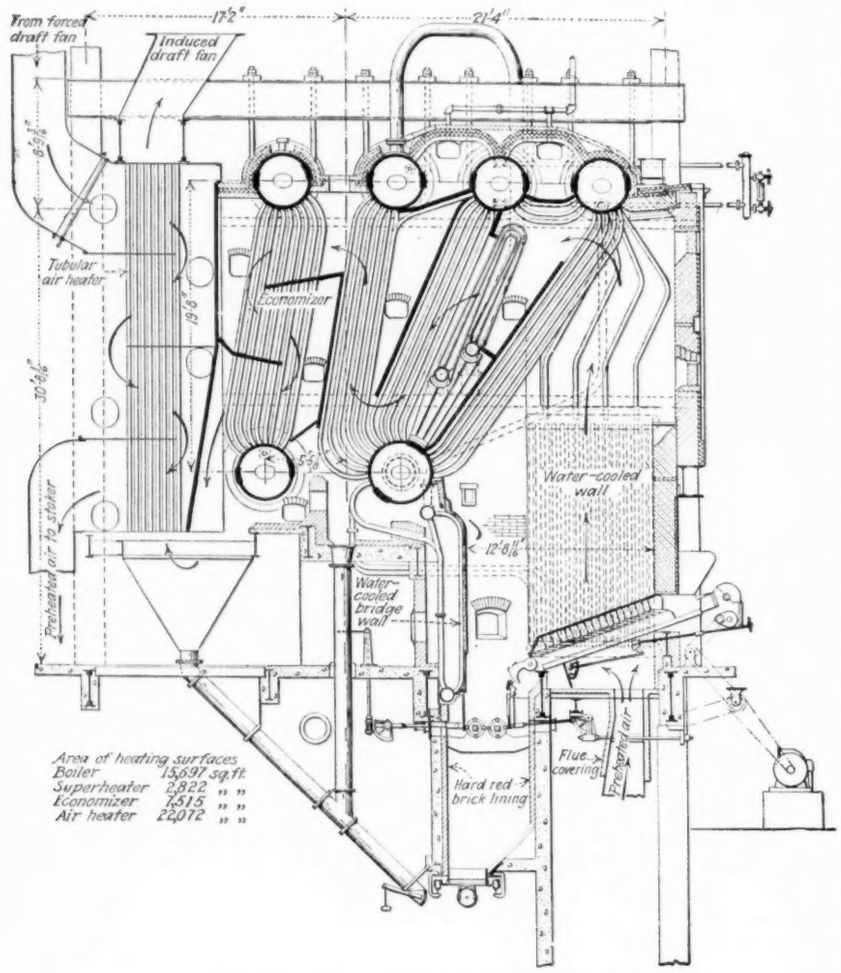


Fig. 3—Cross-section of boiler setting

As previously mentioned, the first two generating units are of 50,000 kw. capacity each. They are single-shaft tandem-compound turbines running at 1,800 r.p.m. In deciding upon the size of unit it appeared that 50,000-kw. machines would best fit into the system of which the new station was to form an important part. The tandem type of unit was selected in preference to the compound unit owing to the greater simplicity of arrangement and to the fact that the smaller space required worked in better with the island-platform plan of foundation and permitted of better ventilation and light in the basement as well as greater accessibility to the main auxiliaries with the crane. The capacity of the two units at present installed can be considerably increased by a slight constructional change at some additional expense.

The low-pressure section of each turbine, which is of the double-flow type, is connected rigidly to a 70,000-sq.ft. two-pass surface condenser placed at right angles to the axis of the turbine. In place of the usual expansion joint in the steam connection or the employment of spring supports, one condenser is carried on counter-weighted levers. This method of supporting the condenser, which was developed by one of the engineers of the company, is designed to distribute evenly the weights of the condenser under all conditions of load and to prevent any change in the loading due to expansion and contraction of the condenser from temperature changes.

The circulating water for each unit is supplied by duplicate motor-driven 54-in. centrifugal pumps each with a capacity of 78,000 gal. per min. The condensate pumps, which are two-stage motor-driven, operating under a total head of 151 ft., are also in duplicate. For air and vapor removal one two-stage air ejector with after-condenser and one hydraulic air pump are provided for each unit.

The arrangement of the intake and discharge tunnels for the circulating water is shown in Fig. 7. The tunnels are carried in below the boiler-room floor, the construction work of the tunnels forming part of the boiler-column foundations. Each tunnel has a connection to one of the pumps on each turbine, so that any one tunnel can be shut down without affecting the operation of any one machine.

The building is of structural steel and concrete. The structural members, with the exception of the roof trusses, are of silicon steel. This permitted the use of smaller members owing to the greater stress allowable, this being 22,000 lb. for silicon as against 16,000 for carbon steel. The roof trusses of the turbine hall as well as the coal-conveyor trusses, were made of carbon steel owing to the large amount of shopwork involved, silicon steel being much harder to drill and work than carbon steel. The adoption of the arched roof, as will be apparent from Fig. 2, in addition to improving the general appearance of the turbine room, materially increases the natural illumination without the use of the usual extensive skylights. The trusses are designed to be self-supporting, no cross-members being used. Any side stress in the structure is taken care of by the boiler house on one side and the switchhouse on the other. The roof of the turbine room is constructed of gypsum with 2-in. cork insulation, the whole being covered with prepared roofing. This type of roof con-

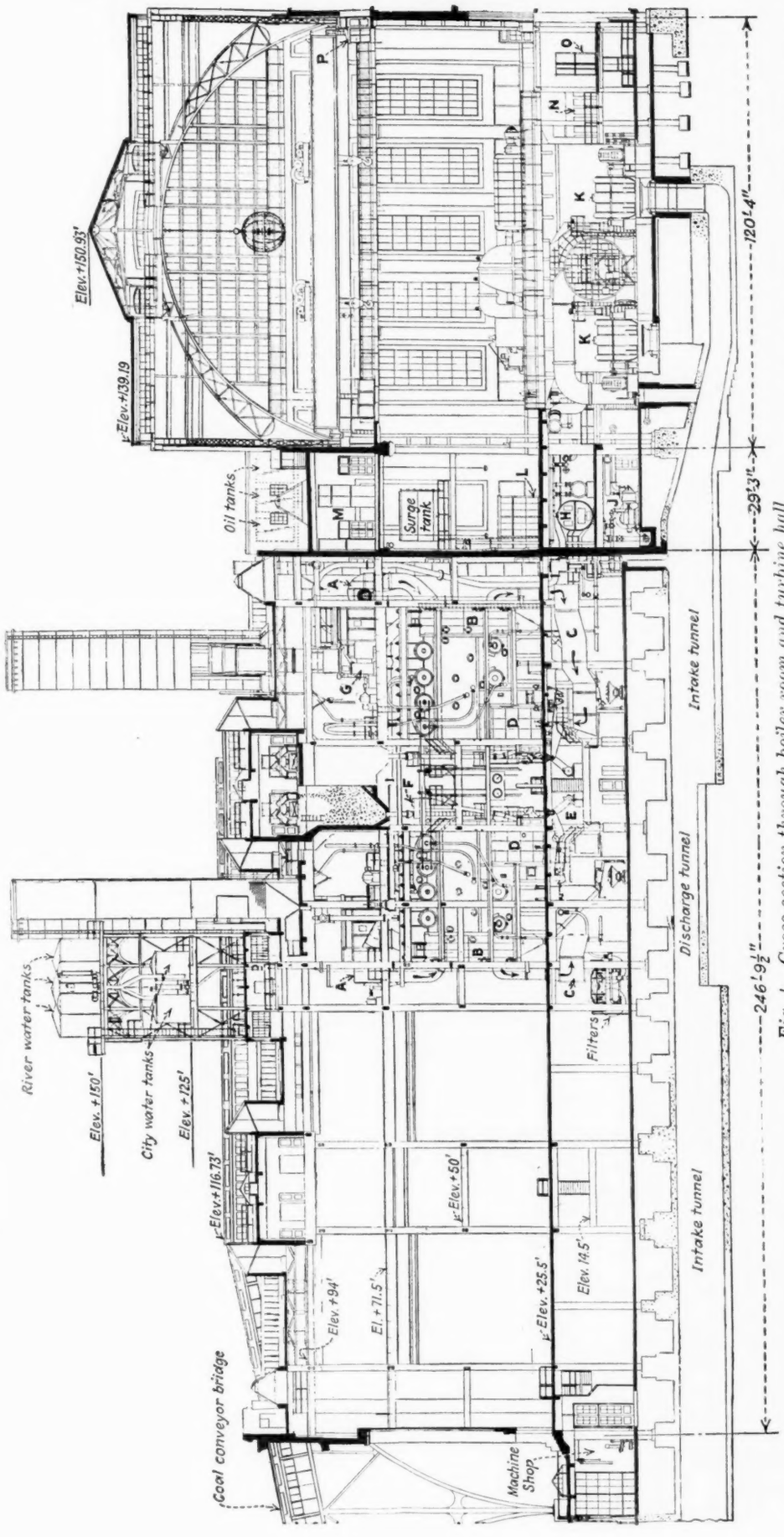


Fig. 4—Cross-section through boiler room and turbine hall

A—Forced-draft fan. *B*—Air preheater. *C*—Duct between heater and stoker. *D*—Water wall section. *E*—Stoker motors. *F*—Scales on test boilers. *G*—Induced-draft fan. *H*—Deaërator. *J*—Feed pump. *K*—Circulating pump. *L*—Shift supt.'s office. *M*—2,300-v. light and power room. *N*—Generator rheostats. *O*—2,400-v. unit aux. bus. *P*—Operating room observation balcony.

struction obviates any trouble from condensation.

In the design of the boiler setting, a cross-section of which is shown in Fig. 3, it will be seen that water-cooled walls have been utilized to probably a greater extent than heretofore for a stoker-fired furnace, the installation on each of the twelve boilers being identical. A large portion of the side walls as well as the complete bridge wall are of water-cooled construction. In the side walls the water-cooled surface is carried to a height about 14 ft. above the surface of the stoker and is also extended below the active furnace area. The upper and lower headers of the water walls are outside the furnace wall proper and are connected to the upper and lower drums of the boiler, as indicated in Fig. 3. The water-cooled surface forming the bridge wall is of similar construction and connected to the same drums of the boiler as the side walls. The water-cooled surface consists essentially of cast-iron blocks faced with refractories and bolted to water-screen tubes which are spaced 6 in. on centers, intimate contact between the blocks and tubes being made by a special filler of high heat conductivity. It is estimated that the water-cooled walls add about 10 per cent to the steaming capacity of the boiler and will materially decrease the maintenance of the furnace walls. The over-all economy of the boiler is also considerably improved owing to the higher CO_2 carried without danger to the furnace refractories.

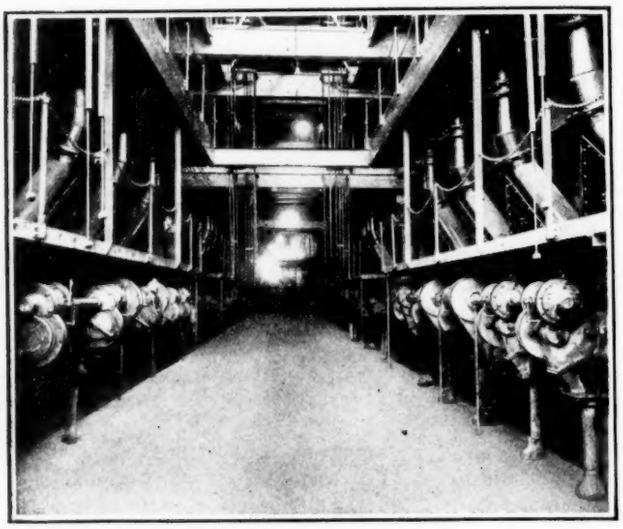


Fig. 5—Boiler firing aisle

Referring again to Fig. 3, it will be seen that the economizer, which has a heating surface of 7,515 sq.ft., is of the two-drum type and is placed parallel with the last bank of boiler tubes within the boiler setting. The tubular air heater is also placed within the same setting, thus giving a very compact arrangement to the entire unit.

Although in some cases the use of both air heater and economizers may not be justified, in the Richmond Station it was estimated that the economizer would give a fair return on the over-all investment and also bring up the over-all efficiency of the station with the limited air-heater surface used. The design and arrangement of the air heater also permitted placing the forced- and induced-draft fans on the same floor above the boilers and the use of short ducts connecting the fans with the air heater and the heater with the wind-box below the stokers. All the combustion gases go through the air heater, no bypass being provided. At 200 per cent of rating and with an inlet air temperature of 80 deg., the temperature of the air leaving the heater runs around 300 deg.

Both the forced- and induced-draft fans are driven by variable-speed motors which are manually controlled through push-and-pull-button controls placed close to the boiler at the operating floor level. The dampers on both forced- and induced-draft are also motor operated, the induced-draft dampers being automatically controlled to maintain a constant draft over the fire.

The underfeed stokers selected are 15-retort, 25 tuyeres long and embody the latest improvements in stokers of this type. Two different makes are installed. The stokers are driven with direct-current motors in order that close regulation might be obtained over a wide range of speed. The motors are on a mezzanine below the boiler operating floor level. The current for driving the motor is furnished from one of the motor-generator units. The total furnace volume is 7,860 cu.ft. per boiler, giving a ratio of furnace volume to boiler-heating surface of 0.496. The boilers are set high, the mud-drum centers being 13 ft. above the operating floor. The stokers are equipped with two-roll clinker grinders, which discharge refuse into an ashpit from which it is dumped through air-operated gates to dump cars and hauled by storage-battery locomotive to a pit. From the pit it is handled by locomotive cranes to barge, railway cars or trucks. The ashpits are of liberal volume and are emptied only once in 24 hours.

Each group of three boilers is served by one brick-lined steel stack 14 ft. 6 in. inside diameter supported on the building. The height of the stacks was made to fit in well with the general design of the building. They are approximately 81 ft. above the foundations. The breechings are designed with a cross-sectional area of 46 sq.ft. for each boiler served.

Coal-handling equipment at the Richmond Station is essentially a duplicate of that used at the Delaware Station. Two towers housing the hoisting equipment are located on piers extending out into the river directly opposite the space between the sections of the station. Inclined-belt conveyors convey the coal from the towers and discharge it to distributing conveyors above the bunkers. The system is arranged so that, when the station is complete and both towers are in operation a shutdown of one tower will affect only one section of the station.

The coal, which is normally received by barge, is first handled by one of two hoisting machines operating 1½-ton clamshell buckets and delivered to receiving hoppers in the tower. From the hoppers the coal passes to picking table conveyors, then through double-roll crushers to inclined flight conveyors. While being carried on the picking tables, all pieces of metal or other foreign materials are removed by an attendant. The inclined conveyors discharge the coal to the distributing conveyors over the bunkers. The entire coal-handling equipment from pier to bunker is in duplicate, either unit having sufficient capacity to take care of the complete station.

Approximately 32 acres of the site, on the downstream side of the station, are reserved for coal storage. This

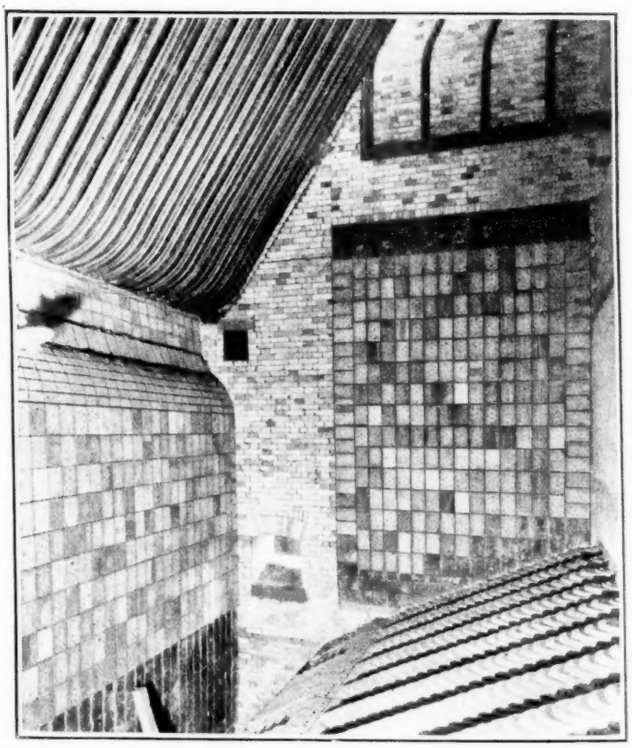


Fig. 6—Interior view of furnace, showing water-cooled section in side wall and bridge wall

will not be used immediately, however, as the main storage yard at Petty's Island, which also serves the Delaware Station, is only about a mile away and has ample storage capacity (250,000 tons) for both stations. As an additional safeguard about three days' supply of coal is held at the dock in barges.

For weighing the coal, as delivered to the station, a weightometer is installed on each inclined conveyor in the base of the tower. No provision is made for weighing the coal fed to each boiler. An approximate check is made, however, by means of revolution counters connected to the crankshaft of each stoker. The amount of coal fed per revolution of the shaft is determined from test on two of the twelve boilers, which are fully equipped with coal scales and other instruments for testing purposes.

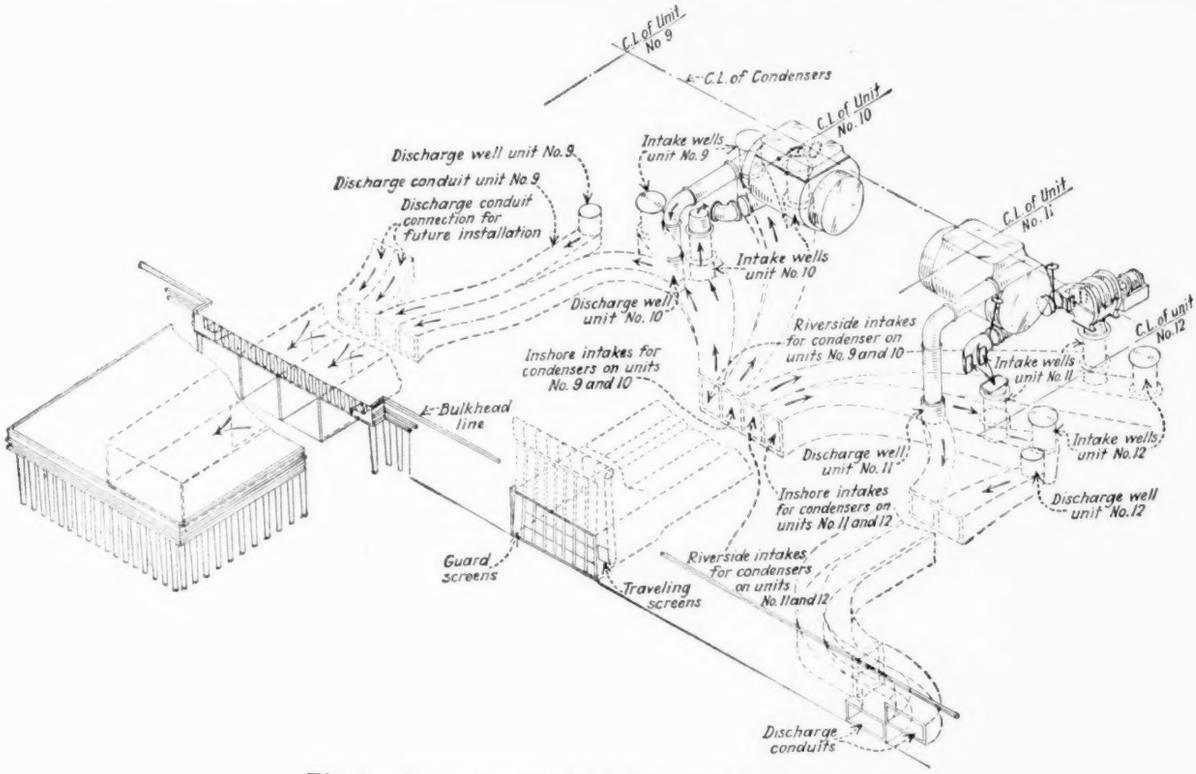

C.L. of Unit No. 9
C.L. of Condensers
C.L. of Unit No. 10
C.L. of Unit No. 11
C.L. of unit No. 12
Discharge well unit No. 9
Discharge conduit unit No. 9
Discharge conduit connection for future installation
Intake wells unit No. 9
Intake wells unit No. 10
Discharge well unit No. 10
Riverside intakes for condenser on units No. 9 and 10
Inshore intakes for condensers on units No. 9 and 10
Intake wells unit No. 11
Bulkhead line
Discharge well unit No. 11
Intake wells unit No. 12
Inshore intakes for condensers on units No. 11 and 12
Discharge well unit No. 12
Guard screens
Traveling screens
Riverside intakes for condensers on units No. 11 and 12
Discharge conduits

Fig. 7—Arrangement of intake and discharge tunnels

All the high-pressure piping in sizes up to 20 in. is seamless steel tubing fitted with forged-steel flanges hammer-welded to the ends of the tubes. When nozzles are used, they are also hammer-welded. All flanges have small tongue and groove with rounded tongue. All bolt material is special chrome-vanadium steel having an ultimate tensile strength of 100,000 lb. The bolts are made in the forms of studs with two nuts, one nut being peened on one end.

It might be recalled that the Philadelphia Electric Co. was one of the first large utility companies to utilize motor-operated valves on high-pressure steam lines. In the Richmond station these valves have been utilized to the fullest extent. In addition to the main boiler and turbine loops, all high-pressure steam and boiler-feed lines are sectionalized with motor-operated valves, the system being arranged so that any section can be cut out without affecting the operation of the plant. The valve motor controls are arranged so that the valves can be closed either from the main station control board or from the local stations, but must be opened from the local stations which are close to the valves. The main control board is in a room off the main turbine hall with convenient entrance from either the turbine or boiler room.

As before stated, the combination of pressures and heat balance finally selected, called for three-stage bleeding of the main units and all electric auxiliary drive. The heat-balance arrangement, as finally worked out, is shown in the diagram, Fig. 8. Leaving the main condenser (see heavy line on diagram, Fig. 8) the condensate passes through the air ejector condenser if in operation to the first heater, which is supplied with steam (at 4 lb. abs., with the turbine operating on a 40,000-kw. load) from the eighteenth stage of the turbine. From this heater it passes to the deaërator condenser and thence to the liquid heater and deaërator, where the temperature is further increased by steam (at 20 lb. abs.) bled from the fifteenth stage. From the deaërator the water goes to the feed pump, which forces it through the evaporator condenser to the third heater and then to the boiler-feed line. While passing through the third heater, the water is raised to its final temperature by steam (at 60 lb. abs.) from the twelfth stage. Provision is made for bleeding the turbine at the tenth stage (95 lb. abs.) also, for heating the building and supplementing the normal supply of steam to the evaporator when an abnormal amount of makeup is required. The evaporator is normally supplied with steam from the twelfth stage. The use of bleeder heaters adds little to the complexity of the station. No regulation is required for the amount of steam bled, this being automatically determined by the temperature and pressure conditions corresponding to the load carried.

The feed pumps, of which there are three for each 50,000-kw. unit, two motor-driven and one turbine-

driven, take their supply of water from the deaërator, which also serves to control the water level of the feed system. The turbine-driven feed pump is for emergency service only and is arranged to come into service automatically in the event of the pressure on the feed line falling below a predetermined point.

As previously mentioned, all the station auxiliaries with the exception of one feed pump per unit are motor-driven, and with the exception of the stokers motors all are alternating current. Motors of less than 50 hp. are served at 550 volts and those of greater capacity at 2,300 volts. Any necessary speed regulation is obtained through the use of brush-shifting or slip-ring motors. All constant speed motors are full voltage starting.

Power for the main auxiliaries is normally supplied through single-phase transformers in a bank connected to the main generator leads. This power can also be supplied from a 2,300-volt auxiliary bus, the change-over being automatic by means of relays. In the event of both sources of power failing, a 2,000-kw. unit is brought into service to supply the auxiliary current. This unit, which is a 2,300-volt 3,600-r.p.m. machine operated non-condensing, is arranged to come to speed ready for load in 15 sec. The unit is started by means of push buttons located at different points in the operating room. Closing any one of the push buttons starts a motor-driven oil pump on the turbine, and when the bearing oil has reached the proper pressure, a regulator in the oil line closes its controls and opens an electrically operated valve in the steam line to the turbine.

The operating room of the station is on the top floor of a connecting building between the switchhouse and turbine hall and looks out upon the latter. The station is tied in with the other three generating stations of the company through 13,200- and 66,000-volt lines.

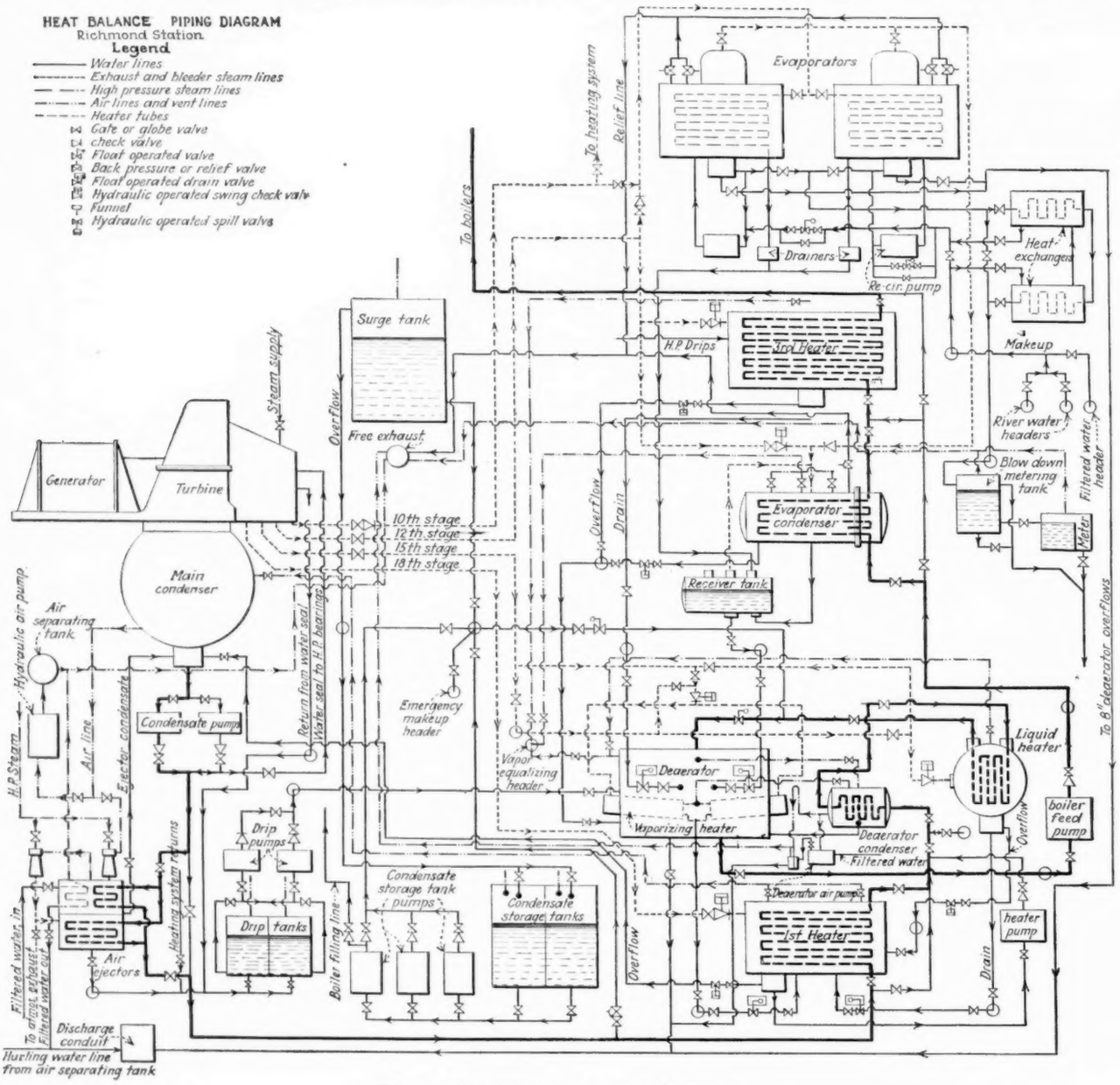


Fig. 8—Heat-balance piping diagram

Heavy line indicates flow of condensate from main condenser through heaters and auxiliary condensers to boiler feed line

PRINCIPAL EQUIPMENT OF RICHMOND STATION

GENERAL

Plant location	Lewis & Delaware Aves.
Capacity installed (1925) kw.	100,000
Capacity ultimate	600,000
Area and volume of first section	Boiler house 79,170 sq.ft. 7,600,000 cu.ft. Turbine hall 47,400 sq.ft. 6,200,000 cu.ft.
Engineering and equipment installation by the Phila. Electric Co. Engineering dept., John T.Windrim, architect; Stone & Webster, building contractors	

IMPORTANT RATIOS

Boiler surface to installed kw.	1.88	
Furnace volume to installed kw.	0.935	
Furnace volume to boiler surface	0.496	
Condensing surface to installed kw.	1.40	
Circulating water gal. per min. to installed kw.	1.56	
Installed kw. to boiler-feed pump capacity gal. per min.	12.2	
Sq.ft. of area and cu.ft. volume per ultimate kw.		
Boiler house (first section)	0.396	38.0
Turbine hall	0.237	31.0
Switchhouse	0.156	14.4

BOILERS AND SUPERHEATERS

Manufacturer	Babcock & Wilcox Co.
Type	Inclined water tube
Number installed	12
Heating surface, sq.ft.	15,697
Furnace vol., cu.ft., 10 boilers	7,780
2 boilers	7,860
Heating surface per cu.ft. furnace volume	2.01 and 1.99
Boiler pressure, lb. gage	400
Total steam temp., deg. F.	707
Make of superheater	Babcock & Wilcox
Type	Triple tube, convection
Superheater surface per boiler	2,822 sq.ft.
Ratio sup. surface to heating surface	0.1797
Furnace water-cooled walls	Bailey Meter Co.
Area in side walls, sq.ft. each	105
Area in bridge wall, sq.ft.	305

ECONOMIZERS

Manufacturer	Babcock & Wilcox Co.
Type	Integral, two drum
Surface per boiler, sq.ft.	7,515
Ratio econ. to boiler surface	0.479

AIR HEATERS

Manufacturer	Babcock & Wilcox Co.
Type	Tubular
Surface per boiler, sq.ft.	22,072
Ratio heater to boiler surface	1.404

STOKERS

Manufacturer (ten)	American Engineering Co.
(two)	Westinghouse Elec. & Mfg. Co.
Type	15 retort 25 tuyere Underfeed
Active area, sq.ft.	Taylor 362 Westinghouse 365
Drive, 15 hp. 250 volt. d.v.	Westinghouse motors
Fuel	Clearfield

DRAFT EQUIPMENT

Manufacturers	F.D. Fans B.F. Sturtevant	I.D. Fans American Blower Co.
Number per boiler	1	2
Capacity, cu.ft. per min.	60,000	45,000
Pressure, ins. of water	9.85	4.85
Drive	G.E. slip-ring motors	G.E. slip-ring motors
Horsepower	125	60

STACKS

Manufacturer	T. H. Livezey & Co.
Number and type, 1 per 3 boilers	Steel, brick-lined
Diameter inside, ft.	14.5
Height, above foundation, ft.	81
Breeching and flues	Connery & Co.

PRIME MOVERS

Manufacturer	General Electric Co.
Number installed	Two—50,000 kw.
Number of stages	20
Bleeder points	Stages 10-12-15-18
Steam pressure at throttle, lb.	375
Steam temperature, deg. F.	675
Water rate at 42,000 kw. 29 in. vac., lb.	9.55 (without bleeding)
Type of generator	3 ph. 60 cyc. 13,800 v.
Exciter	250 kw. 250 v. direct connected
Generator air cooler, sq.ft.	19,686
Ventilating air, c.f.m.	110,000
Cooling water required g.p.m.	9,720
Air cooler fans, four, 55,000 c.f.m.	Green Fuel Economizer Co.

MAIN UNIT AUXILIARIES

	No. 10 Unit	No. 11 Unit
Condensers	C.H. Wheeler Mfg. Co.	Westghse. Elec. & Mfg. Co.
Type	2 pass surface	2 pass surface
Capacity steam per hr., lb.	500,000	500,000
Surface, sq.ft.	70,000	70,000
Number of tubes	12,720	12,734
Diameter, inches	1	1
Kind of tubes	Admiralty mixture	
Tube Manufacturer	Scovill Mfg. Co.	British American Metals Co.
Ratio cond. surface to kw.	1.4	1.4
Circulating Pump	C.H. Wheeler Mfg. Co.	Westinghouse Elec. & Mfg. Co.
Number	2	2
Capacity, each, g.p.m.	78,000	78,000
Head, ft.	19.3	19.23
Drive	500-hp., ind. motor	500-hp., ind. motor
Condensate pumps	C.H. Wheeler Mfg. Co.	Westinghouse Elec. & Mfg. Co.
Number	2	2
Capacity each, g.p.m.	1,200	1,200
Head, ft.	151	151
Drive	100-hp., ind. motor	100-hp., ind. motor
Air pumps	C. H. Wheeler Mfg. Co.	Westinghouse Elec. & Mfg. Co.
Number and type	2 Rado-jet	2 Hydraulic

HEAT TRANSFER EQUIPMENT

Low-pressure heater	Two, Cochrane Corp.
Heating surface, sq.ft.	2,560
Capacity lb. per hr.	450,000
Max. steam pressure, lb. abs.	5.21
High-pressure heaters	Two, Cochrane Corp.
Heating surface, sq.ft.	2,200
Capacity, lb. per hr. with friction loss of 15.3 ft.	600,000
Max. steam pressure, lb. abs.	75.4
Deaerators	Two, Cochrane Corp.
Evaporators	Two, Griscom-Russell Co.

COAL-HANDLING EQUIPMENT

Two Coal Hoisting Sets	Link Belt Co.
Coal Conveyors	Robins Conveying Belt Co.
Coal Scales	
8 automatic (4 on each test boiler)	Richardson Scale Co.
2 weightometers (325 tons each)	Merrick Scale Co.

ASH-HANDLING EQUIPMENT

Ash Gates	R. H. Beaumont Co.
Storage Battery Locomotive	Atlas Car Mfg. Co.

PUMPS

Boiler Feed	Cameron; Ingersoll-Rand Co. Four 1,250-g.p.m. motor-driven, two 1,600-g.p.m., turbine driven.
City Water Pump	Worthington Pump & Machy.Corp. Two 2-stage, 60-g.p.m. motor driven
Condensate storage pumps	DeLaval Steam Turbine Co. One 2,200-g.p.m. turbine driven; two, 500-g.p.m. motor-driven.
Screen Washing Pumps	The Goulds Pumps, Inc. Two 3-stage 500-g.p.m. motor-driven.
River Water Pump	De Laval Steam Turbine Co. Two 3,000-g.p.m.; one 4,000-g.p.m.
Fire Pump	Worthington Pump & Machy Corp. Two 1,500-g.p.m.
Turbine oil storage pump	Viking; W. H. Egan Co.
Emergency oil pump	Viking; W. H. Egan Co.
Heater pump	Worthington Pump & Machy.Corp.

MISCELLANEOUS MECHANICAL EQUIPMENT

Oil purifying equipment	De Laval Steam Turbine Co.
Oil strainers	Andale Engineering Co.
Miscellaneous steel tanks	Coatesville Boiler Works, Lancaster Iron Works, T. H. Livezey & Co.
Air compressor	Ingersoll-Rand Co.
Traveling crane	Niles-Bement-Pond Co.
Revolving screens	Link-Belt Co.
Soot blowers	Diamond; Diamond Power Specialty Corp.
Feed-water regulators	Copes; Northern Equipment Co.
Boiler-feed pump governors	Ruggles-Klingermann Fisher Governor Co.
Water filters	Roberts Filter Co.
Motor controls for valves	Cutler-Hammer Mfg. Co.
High-pressure valves	Lunkenheimer Co.
Stop and check	Schutte & Koerting Co.
Non-return	Foster Engineering Co.
Emergency trip	Schutte & Koerting Co.
Hydraulic-operated valves	Atwood & Morrill Co.
Float-operated drain valves	Atwood & Morrill Co.
Circulating water	Chapman Valve Mfg. Co.
Blowoff	Yarnall-Waring Co.
Safety valves	Consolidated Safety Valve Co.
Atmospheric relief	Atwood & Morrill Co.
High-pressure piping	M. W. Kellogg Co.
Pipe covering	Ehret Magnesia Mfg. Co.
Steam Traps	Armstrong Mfg. Co.
Barge hauling equipment	Lidgerwood Mfg. Co.

Gages, Meters and Instruments: Westinghouse Elec. & Mfg. Co., Bailey Meter Co., Schade Valve & Mfg. Co., General Electric Co. Benjamin Engi Co., Foxboro Co., Yarnall-Waring Co., Brown Inst. Co., Elliott Co., Leeds & Northrup Co., Precision Instrument Co., Uehling Instrument Co., Crosby Steam Gage & Valve Co., Ashton Valve Co., Electric Tachometer Co., J. G. Biddle Co., Asheroft Mfg. Co., A. H. Thomas Co., Paul B. Huyette Co., Casanave Supply Co., L. S. Starrett Co., Elec. Development & Machine Co., Williams, Brown & Earle Co., Cutler-Hammer Mfg. Co., Federal Gage Co., Monroe Calculating Machine Co., Philadelphia Meter Co., James Hogan & Co., Hershey Meter Co., Taylor Instrument Co.s, Fulton Co.

Electrical Equipment:—Westinghouse Elec. & Mfg. Co., General Electric Co., Condit Electrical Mfg. Co., Electrical Development & Machine Co., Weston Electrical Instrument Corp., Electric Storage Battery Co., Q.P. Signal Co., Autocall Co.

Some Problems in the Use of Pulverized Coal

By A. G. Christie
Professor of Mechanical Engineering, The Johns Hopkins University

Rapid development is always accompanied by pressing problems. The author presents suggestions for improving coal preparation and burning, and evidence that ash may aid combustion. He stresses the importance of adequate instruments and the need of a simple means for checking the loss of carbon in flue gas. He advocates the development of dust catchers as a social, if not an economic, necessity.

MUCH has been accomplished in the successful application of pulverized coal to the furnaces of many boiler plants both large and small. Powdered coal has already established an enviable record for high economy in operation and for satisfactory performance. In such a rapidly developing art, however, problems naturally arise which demand the attention of engineers in their solution in order to improve further the art itself. Some of these newer problems are discussed and analyzed in the following paragraphs. The solution of some is already progressing.

Engineers in general recognize that radiant heat is the most effective means of heat transfer in furnaces, yet it appears that its effect in the drying of materials has not always been given the consideration that it deserves. Radiant heat can be an important element, particularly in drying a black body such as coal. If moist coal could be passed in a thin stream under black steam-heated drying coils, radiant heat in large quantities would be transmitted from the coils directly to the surface of the coal and would be conducted from the surface into the coal particles themselves. If a stream of warm drying gases or air were drawn through the coal at the same time, evaporation of moisture would be very rapid. This will be apparent from a consideration of the nature of drying phenomena.

When a substance is dried by passing warm gas or air over it, the gas tends to saturate itself at wet-bulb temperature, giving up a portion of its own heat to evaporate the moisture. Evaporation takes place at wet-bulb temperature throughout the whole drying process, and the substance being dried remains substantially at wet-bulb temperature until dry, when it will naturally rise in temperature, as no more moisture can be evaporated. The drying medium, gas or air, should therefore have a low wet-bulb temperature. This is one of the advantages secured by the steam drier using atmospheric air over the waste-heat drier employing flue gases. When radiant heat is applied, the rate of evaporation is greatly increased, for this heat is added to that given up by the drying gas in evaporating the moisture. Radiant heat is said to have been successfully applied in certain industrial processes for drying other materials, and it is now proposed to apply this latter process to coal.

Driers were not installed in some stations with central pulverized-coal systems and do not appear to be needed with certain classes of coals. On the other hand, the difference between surface and inherent moisture in coals is not well understood, and because of this misunderstanding the failure to install driers has proved a serious handicap in the combustion of some coals. Driers are now more generally installed so as to permit a wider choice of fuels and to insure satisfactory operation under all conditions.

Pulverizing equipment now on the market can be placed in three classifications: (1) Impact mills, (2) centrifugal mills and (3) ball or tube mills. The first two types usually operate at high speed and their parts wear rapidly, hence their maintenance costs are usually high. The third type generally operates at slow speed and should therefore have less wear and tear. However, less development work has been done on the last type than on the others, although these mills have long been in use for pulverizing coal in the cement industry. Many engineers feel that their possibilities have not yet been fully developed. The rod mill has proved a very efficient and satisfactory pulverizer for the reduction of ores at mines. This mill has not, to the writer's knowledge, been tried out on coal pulverization, although he urged its use several years ago.

Much can still be done to decrease both power consumption and maintenance on pulverizing equipment now on the market. This is fully substantiated by a recent paper[1] which details the steps in the mechanical development of a new unit pulverizer and the resulting improvements in performance. Similar development work on other pulverizers, particularly those of the ball and tube mill types, should yield equally satisfactory results. Little information is available on the maintenance costs of unit pulverizers with various coals. This would be valuable information for engineers. Also little information is available on the effect of the ash content of coals on the pulverizing capacity of mills.

Unit pulverizers must be located quite close to the furnaces they serve. Most of these are air swept, and considerable thought should be given to the prevention of possible explosions in these mills when shutting down with a hot furnace. Some arrangements are questionable on account of their liability to back-fire and explosion under certain conditions. These designs should be rectified, for a bad explosion in any of this equipment would discredit unit pulverizers for a long time.

Driers, pulverizers and transport equipment are often located in a preparation house separate from the main boiler plant. This can be amply justified in many locations, such for instance as one where an old building was available for this purpose, or again where the coal

[1] "The Development of a Unit Pulverizer," by R. Sanford Riley and O. Craig. *Mechanical Engineering*, Mid-November, 1925.

will be unloaded and stored at some distance from the power plant and where it can be transported in the pulverized form cheaper than in any other way. Where such favorable conditions do not exist, it is difficult to justify the additional expense of a separate building for preparation equipment. This machinery can usually be housed in a section of the main plant at a lower cost than in a separate building. Objection is often raised that there is danger of explosion, but there is little danger if the ordinary rules of good housekeeping are observed and the pulverizing section is well ventilated. Preparation rooms should be kept scrupulously clean.

The Ash Content of Coal

Large percentages of ash in powdered coal appear to have less effect on the boiler efficiency than in the case of stokers, and very dirty coals have been burned in the pulverized state. However, high ash content of the coal increases the cost of pulverizing and the wear on the grinding and transport equipment; it would therefore be desirable to treat the coal at the mine to lessen the ash content provided this preparation would pay for itself. Freight would be saved and less ash would be discharged from the chimney.

Pyrites is particularly objectionable in coal that is to be powdered and should be removed if a cheap process can be devised. This impurity is often responsible for the rapid wear of the pulverizing mills and of the transport piping; it increases power for pulverizing and also lowers the fusing point of the ash in the furnace. Up to the present no satisfactory process has been devised to remove all pyrites from coal.

While it would seem desirable to remove all ash from coal, there is another side to the question. Dr. R. Lessing, in a recent series of lectures in London on "Coal Ash and Clean Coal," pointed out that the combustion of powdered coal may be greatly influenced by the catalytic action of the ash in promoting rapid chemical union of oxygen and fuel. He shows data that indicate that such catalytic action of the ash is present in the combustion of coal and suggests that this may explain the differences of action in the furnace of coals of similar character, but varying ash. It seems that ash as a catalyst also has some influence on the rate of decomposition of the coal during its coking period. That the ash in the coal may have a catalytic effect beneficial to combustion, will be a new thought to many engineers. This may suggest some new experimental work to determine the true nature and effect of this phenomenon and to indicate which of the various ash constituents has the greatest catalytic influence.

Preheating the Coal

The writer called attention in another article[2] to the desirability of preheating the powdered coal as well as the primary air, before it enters the furnace. This requirement has been questioned on the ground that there is no great gain in heat as a result of this preheating and that it induces coking and sticking of the coal in the burner mouth. Later data, however, indicate a substantial gain from highly preheating the coal.

Tens Rude, in a recent article on "Carbonization of Coal" (*Gas Age-Record*, Dec. 12, 1925), presents experimental data which show that the time of carbonization is greatly shortened by preheating the coal. He points out that even moderately heated air when used to preheat the coal, tends to destroy its coking properties. This principle of preheating the coal to stop its coking tendencies is used in the McEwen-Runge low-temperature carbonization process. Hence it should be possible with the proper method of preheating coal to secure the desired more rapid ignition resulting from the use of highly preheated coal and at the same time avoid sticking at the burner nozzle due to coking. No coking difficulties have been reported at the burners in a plant where air at 500 deg. F. is said to be used, and a short intensely hot flame is claimed. The preheating of the coal as well as of the primary and secondary air seems to be a logical step in development to increase rapidity of combustion and reduce furnace volume.

Furnace Conditions

Combustion must be complete before the flame enters the tubes if the best operating conditions would be secured. Ignition may be slow owing to undried coal, low-temperature primary air, or faulty admission of secondary air. Preheated air tends to remedy these conditions.

The size of coal particle has much influence on furnace conditions. Too coarse grinding of certain coals results in slow and incomplete combustion. Finer grinding may remedy this trouble, but too fine grinding must be avoided. In his little book on "The Dust Hazard in Industry," Gibbs states that sand particles of 0.004 in. radius will settle in still air at the rate of about 10 ft. per sec. while fine smoke particles such as cigar smoke of 0.00001 in. radius will fall only about 10 ft. in 30 days or in fact scarcely settle at all. When powdered coal enters the furnace, it is heated to its carbonization point and later to ignition temperature by the radiant heat of the flame. Combustion proceeds as new supplies of oxygen reach the coal particle through the removal or penetration of the surrounding gases and products of earlier combustion. In other words, combustion proceeds as the coal particle penetrates fresh supplies of air and oxygen. Now if the particle is as fine as the cigar smoke, it will carry along in its own gases and will not penetrate the surrounding gas envelope. This will delay combustion and may even permit fine unburnt carbon particles to pass through the boiler. Very fine grinding may produce such small-sized particles and may therefore be undesirable. Accordingly, attention should be given to the means provided in the various pulverizers whereby neither too coarse nor too fine coal leaves the mill.

It has been pointed out repeatedly that turbulence is a desirable condition in a powdered-coal furnace, to break up the gas envelopes around coal particles and bring fresh oxygen in contact with the fuel, thereby accelerating combustion. The trend of design seems to be toward schemes to secure such turbulence.

In some of the older furnaces with refractory walls, it was necessary to limit the furnace temperature by admitting excess air. When water-cooled walls are used, no such limitation occurs and the highest possible furnace temperature may be carried. Designs have already been prepared with all sides of the furnace water cooled. Secondary air can be admitted to such furnaces by burners provided with mixing devices placed at the furnace and discharging the resultant jets so as to secure the desired turbulence in the furnace. This scheme, to be most effective, should have the secondary

[2] "Boiler Furnaces for Pulverized Coal," *Mechanical Engineering*, August, 1925.

air supplied under a slight pressure, which can be done by the forced-draft fans of the air preheater.

Trouble has been experienced in certain powdered-coal plants with slag on the first rows of boiler tubes, and an analysis of the local factors will generally indicate the cause of the trouble and the remedy.

There are several factors that may contribute to this difficulty. The principal cause is the presence of incompletely burned carbon particles in the flame entering the boiler tubes. This may be due to "short-circuiting" of the flame. That is, the flame or a portion of it, takes the shortest path from the burner to the tubes and enters the tubes while combustion is still in progress. In other cases, improper primary or secondary air mixtures, too short a flame travel or too coarsely ground coal may cause the flame to enter the tubes before combustion is complete. In all these cases the trouble usually begins between the first and second rows of tubes. A hot particle of coal falls on top of the soot on the first tube and, being insulated from the cooling surface, continues to burn. Its ash fuses and forms a collecting spot for more coal particles. The resultant fused ash forms a cap over the tube which grows until it tends to drip between the adjacent tubes. The cooling effect of the tube causes the ash to "mushroom" until the passage between tubes closes over and then the lower side of the tube is soon covered. Once a center of slagging is formed, the trouble spreads rapidly. Certain arch constructions that cause reverberatory effects, or that unduly shield heating surface from radiant heat, or that set up a throat effect with intense after-combustion, tend to aggravate the difficulty.

Generally, a study of the various factors will indicate which one of these causes is influencing the formation of slag. The cleaning of tubes should be a thorough job, for any piece of slag left adhering to a tube may become a nucleus for slag formation when the boiler is again put in service. Clean boiler tubes do not slag, because of the rapid cooling effect of radiant heat transfer.

Control Equipment

One cannot expect the best performance from powdered-coal equipment unless the operators are provided with first-class instruments to indicate operating conditions at all times. Furthermore, the operators must know how to use these instruments and how to control the pulverizers, fans, feeders, etc., in conformity with their indications. This fact should be presented in the strongest possible fashion to owners of industrial plants who are contemplating the installation of pulverized-fuel equipment in small and often poorly equipped boiler plants with the expectation of realizing large savings in coal consumption. Their present equipment of hand-fired grates or stokers often has not a single instrument to indicate performance or to guide the fireman. It seems unreasonable to expect any better performance from the same men with powdered coal without suitable instruments to guide them. Conditions can change so quickly in powdered-coal furnaces that instruments providing instantaneous indications of performance are indispensable. Such an instrument as the steam-flow air-flow meter seems suitable.

Complete automatic control of the whole firing system is another desirable condition, but its cost is high for small plants; in fact, the justification of its expense in large plants has often been questioned. However, it can generally be justified in the large plant on the basis that test conditions can be more nearly approached by automatic than by hand control, and the increased efficiency thereby obtained warrants the added investment.

Carbon in Flue Gas

In the operation of stokers one watches the quantity of the carbon in the refuse, and this is checked periodically in plants where high efficiency is desired. There is practically no carbon in the material that settles at the bottom of a powdered-coal furnace, and hence carbon in such refuse is not checked. There may, under certain conditions, be considerable carbon carried in the flue gases, and this should be periodically checked.

It is not an easy matter to check accurately the carbon in the flue gas. In the first place there is the difficulty of getting an average sample of the flue gas without losing some of the ash that it carries. Even when a good sample is taken, it is difficult to separate the fine ash from the gas and to determine from the flue-gas analysis, the coal analysis, and the ash analysis, the carbon lost per pound of coal. Engineers will welcome a standardized procedure.

Carbon may be carried over with the flue gas when short-circuiting of the flame occurs so that carbon particles entering the boiler tubes before combustion is complete, are chilled below the ignition point. Slow ignition, too wet coal, coarse or exceedingly fine grinding, too low primary air pressure and poor secondary air admission may also be causes contributing to this carbon loss. Sometimes it is difficult to correct the trouble even when it is known to be present and some reconstruction is often needed.

Dust Catchers

The increasing use of powdered coal has created a demand for a cheap, small-sized and efficient dust catcher. Dust of almost impalpable size issues from the vents from certain of the pulverizing mills with central systems. The discharge of driers also carries considerable dust. Bin vents sometimes discharge dust. The presence of such fine coal dust is a nuisance around a plant which every engineer desires to prevent. However, this dust is extremely difficult to catch and reclaim. Water sprays, bag filters, Cottrell electrical precipitators and other types of dust-catching devices have been tried at various places, but all have their limitations due either to high first cost, bulkiness or low efficiency in operation with such dust.

Much of the ash in powdered coal passes up the chimney as a fine powder. This material is simply earthy matter and at present has no commercial value. It would therefore not pay to reclaim it for industrial purposes, but there are social reasons why this dust should not be allowed to pass into the atmosphere, particularly for power plants in large cities. Engineers should therefore exert every effort to stop this discharge of dust up the chimneys. Several devices are being tested for this purpose with promising results, though all are in the development stage. For chimney work the separation should be effected without cooling the flue gas and without too great draft loss.

The foregoing paragraphs refer to problems that may be called the "growing pains" of a new art. Their consideration may suggest suitable modifications in new designs to avoid future difficulties and in present installations to correct these troubles.

Recent Underfeed Stoker Developments

THE underfeed stoker has been used for some twenty years, but its development is still the object of diligent endeavor on the part of its designers. Each improvement is an attack on some definite operating difficulty or thermal loss. This article outlines the losses chargeable to the stoker and shows how recent developments serve to reduce these losses to small proportions.

IN THE DESIGN of many pieces of machinery it is easily possible to determine more or less offhand the proper proportions for various parts, and in examining the finished product an experienced engineer will have no difficulty in developing an intelligent opinion of the merits of the machine through more or less superficial inspection, perhaps checked by simple measurements and calculations.

With stokers, however, this is not possible. The only way to determine intelligently the merits of a stoker is to operate it for a considerable period, and perhaps with a variety of fuels. This makes progress in the development of stoker design necessarily slow, and it is not surprising if now and then a design fails to measure up to expectations. In a stoker in which

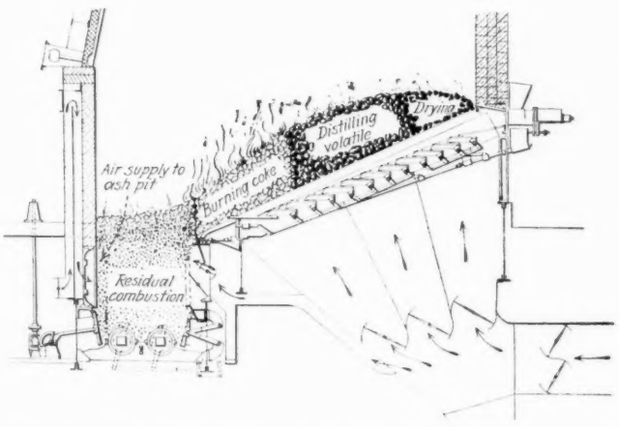


Fig. 1—The four zones of stoker performance are not rigidly bounded, but are sharply enough defined to permit control of the air supply in four parts

the proportions are in some respect not properly adapted to the conditions of operation, it is almost impossible to correct this failure by any modification of operating methods. On the other hand, a stoker that embodies proper design for the conditions under which it is to operate will greatly facilitate proper performance. For example, in the matter of smoke, in certain stoker installations it is actually difficult to produce smoke with the varieties of coal that are usually used.

It is well recognized that a stoker is an apparatus in which the fuel undergoes four distinct processes: (1) Drying out of moisture, (2) distillation of volatile matter, (3) combustion of remaining coke, (4) combustion of residual unburned combustible and the cooling of the ash. These processes take place in reasonably well-defined portions of the stoker structure and a properly designed stoker will so proportion the feeding of coal and air that each of these processes is properly completed in its turn with a minimum of loss.

Characteristic Losses

Each of these four processes is accompanied by characteristic losses. The drying of coal naturally involves the absorption of considerable heat by the moisture evaporated, with the formation of significant quantities

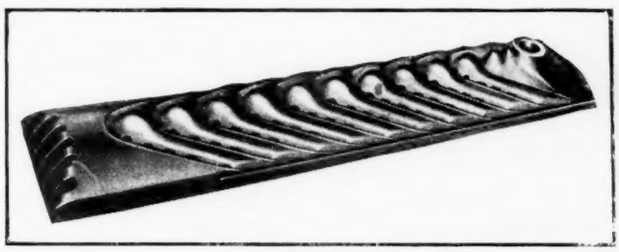


Fig. 2—Air-cooled secondary ram forming moving retort bottom

of water vapor that pass the damper at elevated temperatures and carry off considerable heat. This is a loss which cannot readily be reduced, since there is no feasible method of removing moisture from coal except as vapor at fairly high temperature. Even with an external drier this is true, although with such equipment it is possible to employ waste heat to evaporate the moisture. In any event, the loss is hardly chargeable against the stoker. All that the stoker designer can do is so to promote drying that the process is completed speedily and does not tie up too large a portion of the stoker structure.

The drying zone in a stoker is naturally restricted by anything that facilitates the drying of the coal. Among such influences one of the most important is the use of preheated air, a practice which carries with it other important benefits.

Smoke Reduced by Air Control

The distillation of volatile matter liberates large quantities of combustible gases, some of which may escape unburned. This loss is reduced by proper design of the stoker air control system. With old stokers not embodying recent improvements, it is sometimes desirable to admit air over the fire at the front end of the stoker in such a way as to promote its thorough mixing with the volatile matter rising from the fuel bed. This air should preferably be preheated, and if flue preheaters are not used, a small supply of air may be passed through a hollow front wall or air may be drawn from the top of the boiler room, and admitted to

the furnace through perforated blocks above the ramboxes. Admission of air at this point will often materially reduce the tendency to smoke, a matter of prime importance in most plants.

The combustion of coke is accompanied by two characteristic losses—the escape of unburned carbon monoxide, and the admission of too much excess air to the furnace. A pound of carbon burning to carbon monoxide develops only 30 per cent of its heating value. The remedy for this loss is much the same as that for reducing the loss of unburned volatile matter. The reduction of excess air is a matter of good construction and a design that makes it possible for the operator to control the air supply.

The burning of residual unburned combustible in the refuse may stop short of completeness, with a resultant loss due to unburned solid combustible, for which the remedy is to be found in a better air supply and more time for combustion. If the refuse is ejected while still hot, there is a loss in its sensible heat, but this is small. The principal objection to insufficient cooling of the ash is the increased maintenance on the clinker-grinder rolls and other ash-handling equipment. The remedy is obviously to allow more time for cooling.

Recent Stoker Improvements

Recent developments in the stoker field have consisted of intelligent attacks upon one or more of these losses, all of which are to some extent under the control of the designer, who may vary air distribution, coal feeds in various parts of the stoker, and the dimensions and design of the ash pit.

One of the serious potential losses in the burning of solid fuel is the admission of excess air to the furnace.

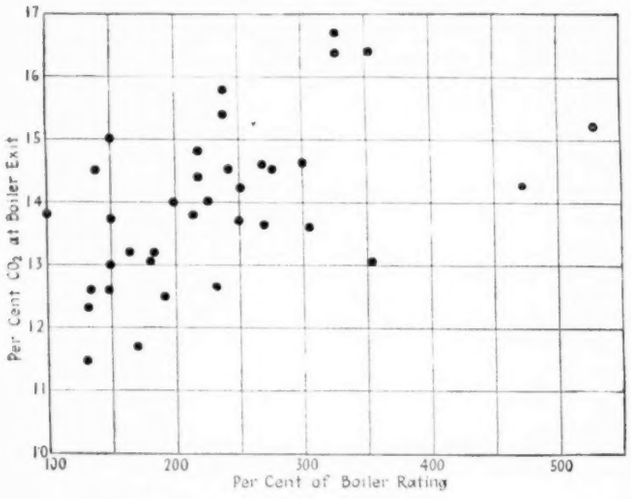


Fig. 3—Control of the air supply reduces excess air to a low value, as indicated roughly by high CO_2 in the exit gas

This may readily be reduced by dividing the air supply into four zones, one for each of the foregoing processes, with separate damper control for each zone. Furthermore, provision should, if possible, be made for thorough mixing of the gases rising from the various portions of the fuel bed, sufficiently early in their path so that combustion may be completed before the gases come in contact with the boiler surface.

Another scheme for improving air distribution is the use of a secondary ram consisting of a short tuyere stack which forms the retort bottom, and has a stroke adjustable from 3 to 6½ inches. This increases fuel burning capacity per retort and gives improved air distribution with reduced air pressure. Low combustible in refuse and low maintenance are claimed.

The success achieved in reducing the loss due to excess air in stoker operation is indicated by Fig. 3, which shows the percentage of CO_2 actually obtained in

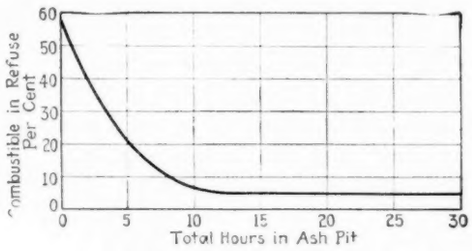


Fig. 4—The longer the refuse remains in the ash pit the more completely is it burned out, to an apparent limit of about 5 per cent combustible at twelve hours

seven modern stoker plants at various rates of steaming. The highest values have naturally been obtained with preheated air, but in a number of plants using no preheat very creditable performances have been achieved. Some of these points correspond to the best boiler and furnace efficiencies thus far recorded.

A noteworthy development has been in the recognition of the importance of ash-pit design and the application of experimental results to the design and construction of actual stokers. This constitutes an attack upon the loss due to unburned combustible in the refuse whereby this loss has been reduced to the order of 0.5 per cent of the heating value of the coal, through a reduction of the percentage of combustible in the refuse from a value ranging from 10 to 20 per cent, or even more, to a figure somewhat less than 5 per cent. This performance represents an approach to complete combustion that can hardly be excelled by any known method.

The deep ash pit, with rotary clinker grinders, was introduced in underfeed stoker practice 15 or 20 years ago, and has since been in successful use. However, it has been customary to look upon this device as primarily a mechanism for the continuous removal of ash and clinker from the furnace without the disturbance to firing and the admission of excess air involved in the use of dump plates or other earlier devices. This is indeed a valuable feature of the rotary clinker grinder, but it has an added function in promoting the efficiency of the combustion process which is commonly overlooked. An examination of patent claims running back for ten years or more indicates that this function of the clinker grinder was clearly recognized by designers, who have been continuously developing the device in the direction of improved furnace efficiency.

Proper Air Supply Essential

In order that the unburned combustible in the refuse shall be reduced to a minimum, it is necessary that air be supplied in adequate quantities to support the necessary combustion and that the admission of too much air to the furnace be strictly avoided, lest the loss due to excess air be increased to counteract the gain due to

reduced combustible in the refuse. Dampers must be provided in the duct supplying air to the ash pit. Recent experience indicates that the admission of air somewhat preheated is very effective in reducing the combustible in refuse to a low figure, and for that reason designs have been prepared whereby the back wall of the furnace is cooled by air on its way to a perforated casting in the face of the bridge wall, which forms one side of the ash pit.

Another obvious requisite for the reduction of unburned combustible in the refuse is that the material shall remain in the ash pit for a sufficiently long time for the completion of this combustion. Inasmuch as the combustible in the refuse is only a small proportion of the total mass, it burns very slowly and requires not only the encouragement of warm air but also a long time for its completion.

Long-continued tests indicate that there is a definite relation between the residual unburned combustible in the refuse and the length of time that the refuse remains in the ash pit above the level of air admission. For short time intervals—that is, for small ash pits or rapid rates of feeding—the unburned combustible is a large percentage of the refuse. As the time interval is made longer by slowing the rate of feeding or by greatly enlarging the ash pit both in cross-section and in depth, the percentage of combustible in the refuse decreases rapidly with increase of time interval. After a fairly definite period is reached, however, further increase of the time does not result in any material reduction in the unburned combustible. Apparently there is a minimum percentage of unburned combustible beyond which it is impossible to go no matter how long the time of exposure to air and heat.

This information, carefully worked out experimentally by the stoker designer, makes it possible for him to lay out a proper ash pit for any given operating condition. This ash pit is designed of a size such that when passing material at the maximum expected rate, the time spent by the refuse in the ash pit is sufficient

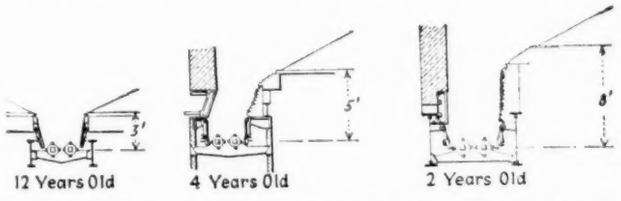


Fig. 5—The growth of ash pits during the last twelve years has been striking

to reduce the unburned combustible to substantially the minimum value. When the stoker is operated at reduced ratings, the refuse remains in the ash pit for longer periods than are actually necessary, but this introduces no difficulties whatever.

In a general way the width and depth of the ash pit vary with the length of the stoker. No rigid relations prevail, however, because of variations in the nature of the coal that may be used. Stokers must obviously be designed and chosen with regard to the service they are to perform.

Operating Methods

It seems obvious that in operating a stoker of this sort the fireman must carry a high ash pit—that is, he must maintain the ash pit full of refuse at all times. To this end, it has proved desirable if not essential to run the clinker grinder rolls continuously at a speed adjusted to the prevailing rate of feed of the stoker. A wide range of speed is easily attained through the ratchet type of drive commonly used, sometimes in combination with a two-speed gear box or a suitable hydraulic pump and cylinder mechanism. It is possible

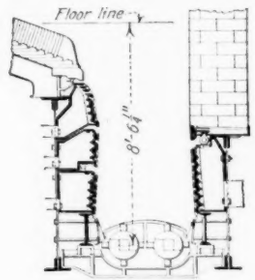


Fig. 6—This ashpit design gives excellent air distribution and control and prevents the possibility of ash arching and hanging in the ash pit

to reduce the speed to a point at which the rolls rotate one revolution in seven or eight hours.

It requires some effort to persuade firemen to carry a high ash pit, but it can readily be demonstrated that such operation is practicable and economically advantageous, except perhaps when burning unusually bad clinkering coal at high rates of firing. The modern underfeed stoker is able to burn coal which would have been impossible for older designs. Save under the most exceptional circumstances, the method of operating with a full ash pit is proving entirely practicable.

It is interesting to note the history of stoker design in this particular. Fig. 4 shows progressive ash-pit designs in the last decade, drawn to the same scale. The depth of the pit has increased from 3 ft. to 8 ft., the width growing approximately in the same proportion. Of course at the same time the size of the stoker has increased and with it the rate of feeding fuel and hence the rate at which ash enters the ash pit. Accordingly it is not proper to assume that the length of time a given particle spends in the ash pit is in direct proportion to the size of the ash pit. It is, however, true that the time spent in the ash pit by a given particle of refuse has increased materially, doubtless to two or three times the figure prevailing ten years ago.

In recent designs the clinker grinder roll has been made much larger than formerly, especially for large stokers and for high ash coal. This makes the bottom of the ash pit wider and seems to give better operating results.

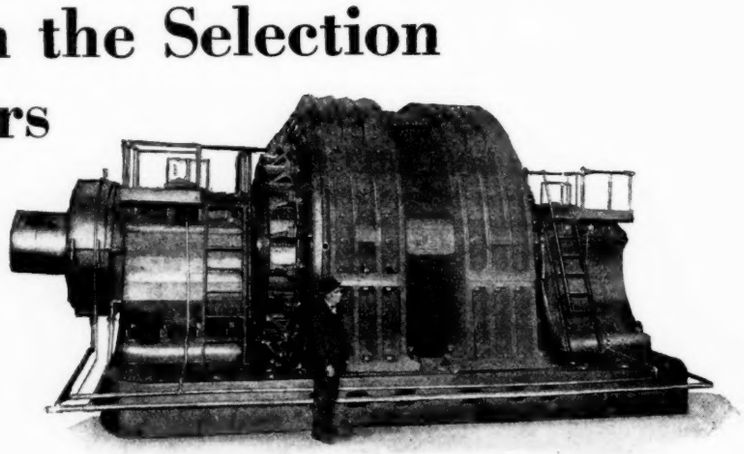
Although this may appear to be a small and simple change in stoker design, it has taken a number of years of careful study and reflection to develop it, and it has proved to be very effective in reducing the particular loss against which it is directed.

It is through precisely such small steps that the development of devices of this sort continually approaches the perfection at which engineering practice continually aims. Stoker development is going forward today faster than ever before. Study of operation defines the problems, and the ever-increasing skill of the designer speeds their solution.

Considerations in the Selection Of Electric Motors

BY J. ELMER HOUSLEY

THERE are many factors that influence the selection of a motor and the kind of drive used. In this article the author considers this problem from the broad economic aspect rather than the individual application.



Double direct-current motor rated at 7,000 hp.

IF YOU were asked to name the universal motor, would it be a squirrel-cage induction type, operating on three-phase 60-cycle current at 440 volts? The requirements of industry and the adaptability of electric equipment militate against the adoption, exclusively, of the predominating motor just mentioned. For moderate-sized plants such a system has much to commend it. In analyzing the application of motors we should proceed from the available source of power, the voltage, frequency, phase and capacity of lines as related to voltage drop at various demands. Next, the area to be covered by the distribution lines to the motors influences the voltage, if a change is possible, and in turn the amount of conductor. The size of the plant, quality of labor employed, number of electricians employed or desired should influence the selection of the type of motor and control to be used, it being assumed that no unusual economic problems are present.

The question of speed for the various motors will be influenced by size and tension of belts, speed reduction necessary, type of drive or class of machines to be driven, with many compromises advised to secure a small number of different speeds to permit interchanging of motors in case of trouble and to reduce number of spares required. In explanation of the influence of size of plant and labor conditions, it will be found that a moderate-sized plant with a small maintenance crew requires a simple installation using squirrel-cage motors because of the simple construction and small amount of maintenance required by such equipment.

The next group considered is the larger plant or small plant with unusual conditions, and such plants may be expected to have a sufficient number of special drives to require two power systems or to justify a maintenance crew to care for motors of a "special" nature or that require skilled supervision.

A study must be made of the area to be covered and the size of motors required in order to determine if a portion of the larger motors may operate at higher voltage, say 2,200 volts, or if the plant must be divided into "load centers" with group low-tension distribution. Only in such a plant would a direct-current system for special drives be considered, and where these drives are closely grouped, then the losses and benefits should be considered for the group and not for the individual drives.

The requirements of adjustable speed alone, at the present time, do not necessarily justify direct-current motors unless close regulation and constancy of speed at any setting are essential or else some advantage is to be gained such as a straight-line variation of current with load to give closely regulated relay protection. Where only a few drives require speed regulation, often a slip-ring or polyphase brush-shifting motor would offer the best solution.

Generalities may be misleading, so when treating the problem of special direct-current applications, they may be grouped into three classes having respectively a maximum demand of 25 to 75 kw.; 75 to 500 kw. and 500 to 2,500 kw. Based on studies covering installations in service, the first class was served by a 125-hp. induction-motor direct-current generator set. The second group, in a particular instance, indicated that a 225-kw. generator and 300-kva. synchronous motor combination offered the best solution. The direct-current load consisted of direct-current trolley industrial locomotives, cranes and close speed-regulated motors. The remainder of the plant power requirements consisted of a 1,500-kw. load of squirrel-cage motors with a power factor of about 75 per cent which the synchronous motor generator set materially improved.

In the third group an installation having a demand of 2,000 kw. from heavy direct-current mill motors, requiring close speed regulation, a satisfactory installation was made using a 2,500-kw. rotary converter. The power factor of the rotary is ordinarily kept between 85 and 90 per cent by properly adjusted transformer taps and regulating the field rheostat of the converter. The mill auxiliary motors are squirrel-cage or variable-speed alternating-current type. This permits operation of shops and auxiliary service on holidays without incurring the losses in operating a large rotary which would be equal to or exceed the auxiliary load required.

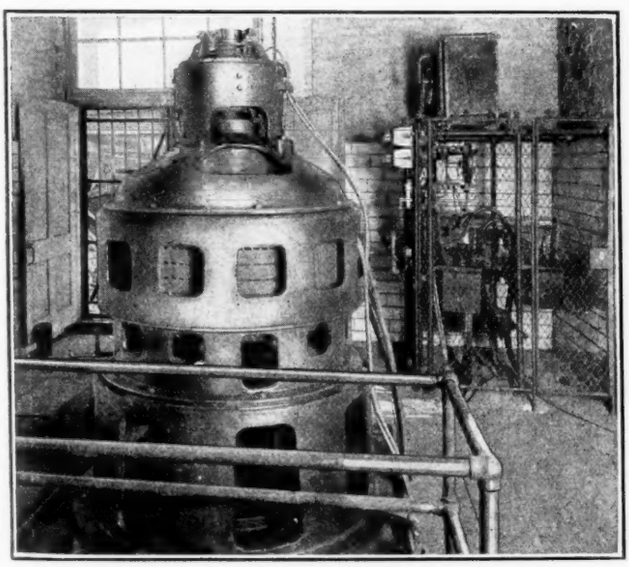
Passing to the mooted question of individual versus group drives, we find that the trend is toward the individual drive in process work in the average industrial plant. Certainly, when considering a group such as eight unit drives from a lineshaft operated by a 200-hp. motor, a considerable delay would be caused by a half-hour shutdown to splice a belt; in other words, there would be lost four unit hours. After all eight belts in

the group had been spliced 8 × 4, or 32 unit hours would have been lost. With individual drives the corresponding loss would be four unit hours. This will indicate the economic advantage in the face of slightly increased motor losses and decreased power factor; however, the latter may be counteracted by several methods.

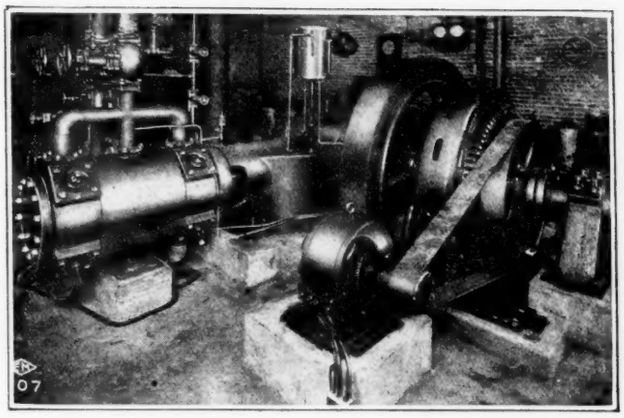
In industrial-plant machine shops the situation is slightly different, but in a majority of instances the group drive is confined to a few machines that may be classed as permanent and having a high percentage of time in operation. Machines of recent manufacture are usually designed for individual drive, and the drives are more easily guarded or may not require guarding at all, while older machines are sometimes equipped with such a drive only at considerably increased expense for proper speed-changing equipment and to meet present-day requirements for safety guards. Conversely, it is often an economic advantage to use a direct drive for a modern machine rather than supply a large amount of shafting, hangers, belts and guards, whereas the cost of the guard alone may sometimes exceed the cost of a motor necessary to drive the machine direct.

Machine tools are precision equipment and in consequence expensive. This involves the question of the advisability of overmotoring such equipment by a large group motor. The possibility of machine and tool breakage is diminished by an individual motor carefully selected to furnish the power demanded by the machine and protected by some relay device with very little time delay characteristics which will shut down the motor due to overloads which would stall the machine or damage the equipment. The general layout of the shop in regard to assembly and fitting may either preclude or indicate the use of group drive, and this factor must be considered.

Another phase of the matter is the lighting problem. Where artificial illumination must be used, the group drive may interfere with the proper placement of the lighting units whether of the incandescent or Cooper-Hewitt type. This interference can be remedied at less extra expense if it is handled as a co-ordinated problem when the shop is designed. Where a shop is engaged in production work rather than job work, the group drive may offer distinct advantages in power consumption, better power factor, fewer motors and reduced motor maintenance.



Vertical synchronous motor driving a 2,000-gal. per min. deepwell pump



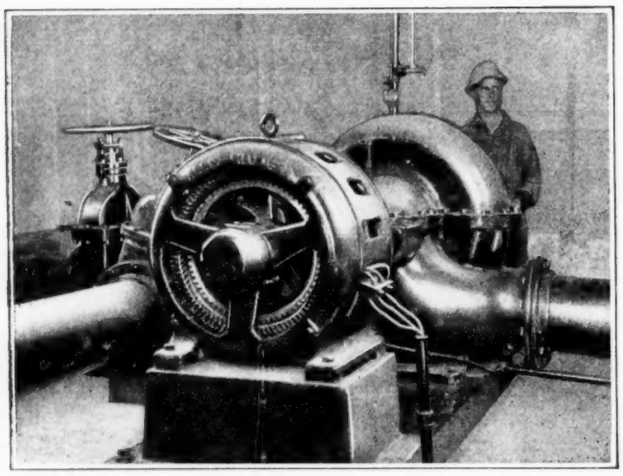
Synchronous motor directly connected to ammonia compressor

In the job shop or industrial plant shop a rearrangement of machines may sometimes be necessary due to change in class of work handled or in the type of machinery in use; in such cases there is a chance for forethought in planning the motor drives. The direct drive permits the utmost in flexibility of shop layout and future alteration in location can be made with little inconvenience, especially where the wiring of the shop has been installed with such an idea in mind. The wiring mains may follow the crane girders on both sides of the shop with outlet boxes at each bay of the shop. Switches and control may be mounted against the columns supporting the crane runway, and ordinarily the machines will require only a short conduit to the motor.

Classification of Motors

Summarizing the application of motors, we must appreciate the cardinal points of any installation; continuity of operation, reliability and the safety of operators and equipment. The construction and insulation of the motors must be adapted to the conditions of operation. Where moisture or dust is present, the motors must be protected in accordance with the nature of the material as regards abrasion, current leakage and mechanical damage. Moisture and chemicals, when present, require insulation which will resist the action of the injurious matter. Bearings are furnished in various types to meet nearly all conditions of dust and loads. Temperature ratings of the motors may well be corrected to meet the load and ambient temperature in which they operate so that a safe temperature of the windings will not be exceeded.

Full-voltage starting of squirrel-cage motors is now being adopted in a few plants for certain horsepower sizes, depending on speed, type of coils, construction of rotor and number of times the drive is started in a given period, also the nature of the drive. Some of the advantages are simplicity, automatic restarting, compensator elimination and low cost of installation. To offset the gains the starting current is increased to as much as ten times normal in some cases, severe vibration of coils in starting, shock to driven equipment and the fact that each installation should be referred to the



Squirrel-cage induction motor driving a single-stage centrifugal pump

maker to determine the suitability of the motor for this condition. The double-squirrel cage type is often recommended.

The various motors made up in standard lines are generally classified as constant speed; varying speed; multi-speed motors; adjustable speed; and adjustable varying speed. The last four are defined by the A.I.E.E. standards as follows:

A varying-speed motor is one whose speed varies with the load, ordinarily decreasing when the load increases.

A multi-speed motor is one which can be operated at any one of several distinct speeds but which cannot be operated at intermediate speeds.

An adjustable-speed motor is one in which the speed can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load.

An adjustable-varying-speed motor is one in which the speed can be varied over a considerable range at any given load but when once adjusted varies with the load.

Some of the constant-speed motors are the direct-current shunt, alternating-current synchronous, squirrel-cage and wound-rotor types. These types are all in general use.

The varying-speed type is the direct-current series. This motor is usually applied to traction and crane service and fans or other positively driven loads.

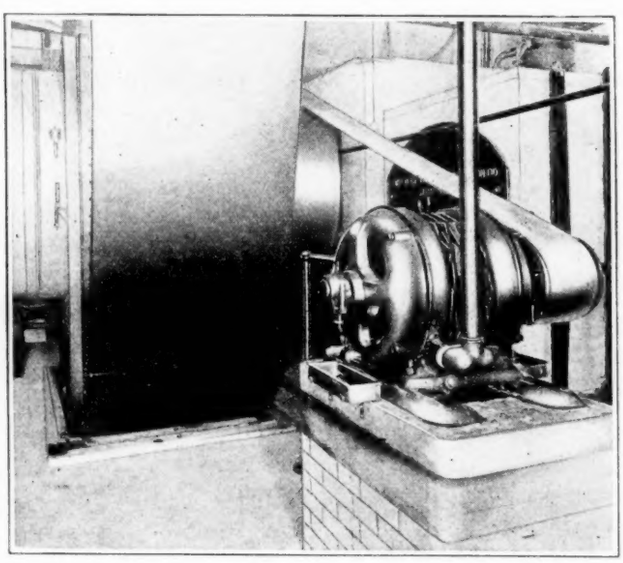
The compound direct-current machine is in a class between the series and shunt types. Its speed characteristics may be practically the same as a shunt motor if the series winding is light. On the other hand, if the series winding is heavy the speed may vary over quite a wide range with variation in the load. A series winding of the proper proportion connected in opposition to the shunt winding will give the motor even a more constant speed than the shunt type. The motor is used on applications that have high static or high friction loads at starting, such as machines with heavy flywheels or worm-geared elevators.

The multi-speed motors are typified by the alternating-current pole-changing type, either squirrel-cage or slip-ring. For a 50 per cent speed range a series-star connection for low speed and a parallel star connection for high speed are sometimes used. Two separate windings may be used and the number of poles may be changed in order to obtain different speed changes.

The adjustable-speed motors are the direct-current shunt-wound field-controlled type; direct-current separately excited with rheostatic-control; direct-current motors separately excited with adjustable or multiple armature voltages; and alternating-current brush-shifting motors with shunt-motor characteristics. These motors are well known except possibly the brush-shifting alternating-current motor. This motor has its primary winding on the rotor and the secondary winding on the stator; a "regulating" winding is placed in the rotor slots also and is connected to a commutator, while the primary goes to a set of slip rings. The phase-wound stator is connected to the brushes so that each phase is short-circuited when the two sets of brushes are in line, in which case the machine becomes an induction motor. Shifting the brush yokes in opposite directions, brings into action a part of the regulating winding whose voltage effects the change of speed.

Adjustable varying-speed motors may include alternating-current wound-rotor with secondary resistance, also with pole-changing devices, and brush-shifting motors with series characteristics; direct-current shunt-wound motors with series armature resistance, compound-wound with field control or armature control, and series wound with armature control. Most of the direct-current motors of this class are for rather special applications and are sometimes difficult to operate and design, owing to the necessity of the torque having to be predetermined. The alternating-current slip-ring motor is extensively used for pumps, fans and hoists where a speed range of 50 per cent is satisfactory and high starting torque is desired. These motors cannot be used on loads where the series characteristics of speed variation with load are undesirable.

The alternating-current brush-shifting motor with series characteristics has an armature similar to a direct-current machine. The windings are connected in series through the brushes with the secondary of a transformer to reduce the voltage within commutation limits. The stator of the motor is similar to an induction motor and is in series with the primary of the transformer. The motor behaves like a slip-ring motor, but has a higher efficiency at part loads and has a wide speed range of infinite steps. It is considerably more expensive to purchase and maintain. However, it is often economically justified for fan and other drives.



Squirrel-cage induction motor driving a large ventilating fan

Superheat and Reheat

By B. N. BROIDO
Consulting Engineer, The Superheater Company, New York City

THE author reviews the reasons for using superheat and reheat, discusses design problems, showing that the superheater must withstand severe operating conditions, and describes three schemes for reheating low pressure steam.

THE important improvements in plant economy of the last few years have been due largely to the use of higher steam pressure and temperature. High initial superheat reduces the moisture carried by the steam through the latter stages of the turbine and thereby reduces friction and windage losses and cuts

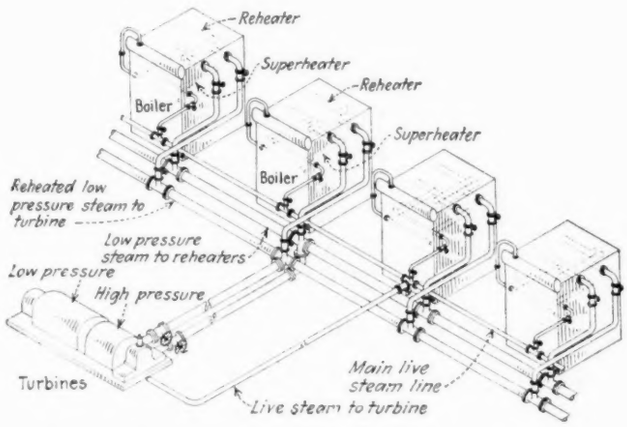


Fig. 1—With a reheater in each boiler setting an extensive system of large piping is necessary

down the erosion of buckets and nozzles. This has a bearing on maintenance expense and on operating cost, since worn blading lowers both the efficiency and the capacity of a turbine.

The higher the initial steam pressure, the higher must the temperature be to produce a satisfactory reduction in condensation in the lower stages of the turbine. For pressures above 500 lb. it is advisable to resort to reheating; that is, to take the steam from the turbine after its pressure and temperature have been reduced considerably, and reheat it to a high temperature so that it will remain dry while passing through the lower stages of the turbine.

It has been recognized for a long time that whatever gain would result from reheating would in the main be due to the increase in turbine efficiency resulting from the reduced moisture content of the steam in the lower stages, rather than through the improvement in the cycle. The heat consumption of a turbine installation may be decreased from 6 to 7 per cent as a result of re-superheating the steam.

There seems to be a rather broad pressure range within which re-superheating may show approximately the maximum saving. Re-superheating more than once has also been given consideration. Calculations indicate, however, that taking into account the pressure drop in the re-superheaters, and the increased invest-

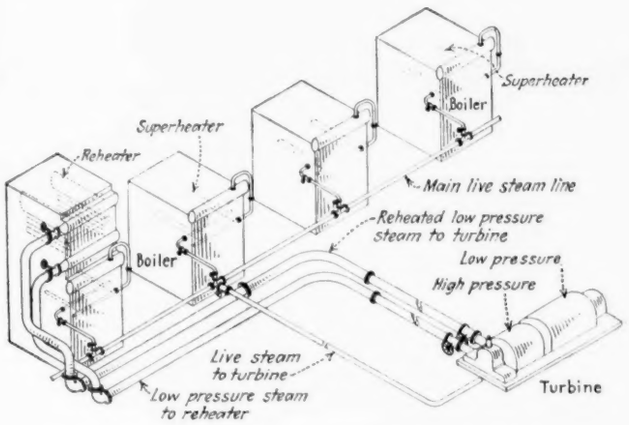


Fig. 2—With a single reheating boiler the large piping is only slightly reduced

ment, complication, etc., the gain possible by going through more than one stage of reheating would be comparatively small.

The initial steam temperature is limited by the materials now available for the construction of equipment. In this connection, one point is often overlooked—of all the parts of a power plant the superheater is exposed

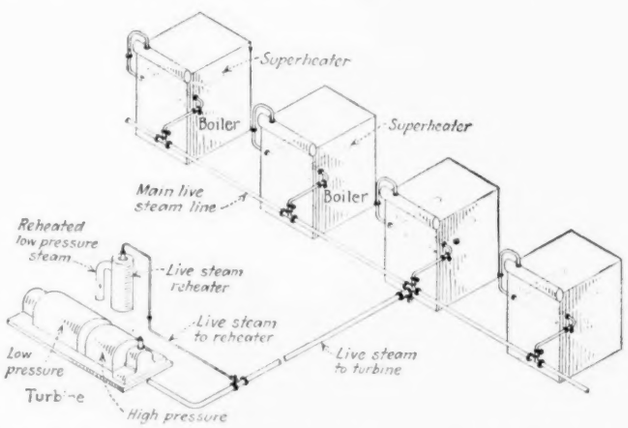


Fig. 3—The steam-heated reheater uses little large piping and gives a compact layout

to the most severe conditions. It carries the steam at the highest temperature and is at the same time exposed to gases at still higher temperatures. The limitation for a superheater is, therefore, not the steam temperature, but the temperature of the tube wall exposed to the gases. From this point of view high pressures are more desirable than high temperatures.

The higher the pressure, the denser is the steam, the greater is its cooling effect on the tubes and the nearer is the tube metal temperature to that of the steam. For the same steam and gas velocity and the same final steam temperature, a superheater tube is better protected from overheating by the higher steam pressures.

The seamless open-hearth steel tubes now used in certain designs can be applied safely for steam temperatures of 800 deg. F. There are now superheaters furnishing steam at these high temperatures with very little fluctuation at the various ratings. This is accomplished by so locating the superheating surface in the gas path that the heat absorption by the boiler and that by the superheater vary in approximately the same proportion. Temperatures of 750 deg. and more can be adopted, since one can be sure that this temperature will not be greatly exceeded. These temperatures can be used with any pressures at present considered advisable, up to 800 deg. and 1,000 lb. per sq.in.

For the higher pressures smaller diameter tubes are used, so that the steam velocity keeps the temperature of the tube walls within safe limits.

Reheating

Four stations in this country and one in England have demonstrated the entire practicability of reheating, and tests have borne out the expectations of improved plant efficiency. However, since the superheating of live steam usually takes place within the boiler setting and by the boiler gases, it was generally thought that

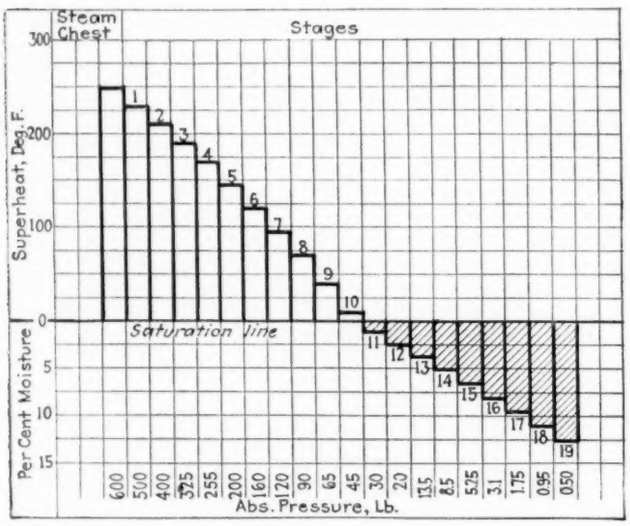


Fig. 4—Without reheating, excessive moisture impairs the efficiency and increases wear in low-pressure stages

this would also be the best method for reheating steam. This is not necessarily true.

There has been developed another method of reheating which is simpler and appears to have other important advantages. This consists of heating the steam from an intermediate stage of the turbine by live steam from the boiler. As the live steam is at much higher pressure, all its latent heat can be utilized at a considerably higher temperature. A comparatively small amount of live steam is required for reheating. With 1,000 lb. of live steam about 20,000 lb. of low-pressure steam can be superheated 100 deg. so that only 5 per cent of the steam used in the turbine will be required for reheating the low-pressure steam 100 deg. If the reheater is properly insulated, its efficiency is near 100 per cent, since all the heat taken from the live steam is imparted to the low-pressure steam. The live steam condenses in the reheater and the condensate is returned either to the hotwell or directly to the boiler without

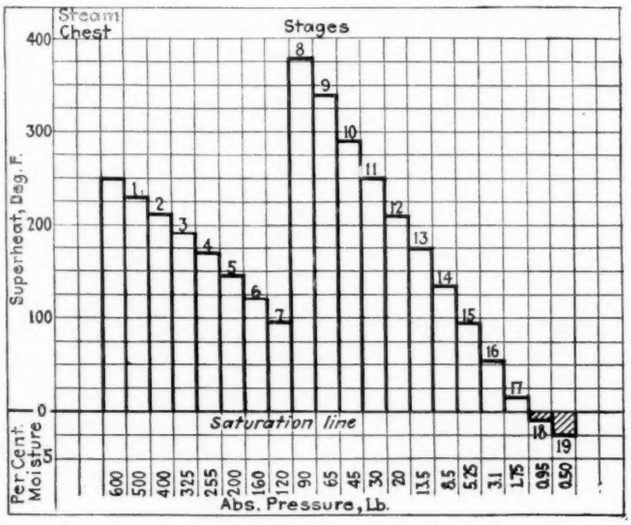


Fig. 5—High-temperature reheating almost eliminates the moisture

heat loss. The reheater is placed near the turbine so that long and large pipe connections between the turbine and the boilers are eliminated.

Owing to the fact that the heat absorbed is the latent heat of the live steam, which is very easily given off, the heat transmission coefficient in the live steam reheater is high.

The outstanding advantage of this method of reheating is the simplicity of regulation, since all that is necessary is to adjust a comparatively small valve admitting live steam to the reheater.

With a reheater placed in the boiler setting, if the turbine is suddenly shut down, the reheater tubes are endangered by the gases in the boiler and by the heat accumulation in the brickwork. This is a serious disadvantage of a boiler gas reheater. The maximum temperature, however, which occurs in a steam-heated reheater is that of the superheated live steam, which temperature cannot have any ill effect on the reheater tubes even with a sudden interruption of the low-pressure steam flow.

There is another advantage of this method of reheating as compared with others. With reheaters in the boiler setting, a considerable amount of low-pressure steam is contained in the large piping to and from the reheater and in the reheater itself. If, for some reason, the load on the turbine drops suddenly and the turbine governor valve closes, this low-pressure steam would be sufficient to cause the turbine to run away, except for low-pressure regulators or automatic valves which complicate the whole arrangement considerably. If the steam is reheated by live steam, the pipe connection is short and, owing to the possibility of crowding the surface in a small space, the volume of the reheater itself is also small, so that there is no danger of overspeeding the turbine and no special regulators are required.

As a matter of fact, the valves between the turbine and the reheater can be omitted. If for some reason, no reheating is required, or if the high-pressure coil in the reheater is defective, all that is necessary is to close the valve in the live-steam supply line and the turbine can operate with the steam passing from the high-pressure to the low-pressure turbine through the reheater.

Figs. 1, 2 and 3 are diagrammatic illustrations of the three methods of reheating. Fig. 1 shows an arrangement of boilers and turbines with reheaters installed in each boiler setting. Fig. 2 illustrates an installation with one large reheater in one of the boiler settings. In both cases large pipe connections between the boilers and turbines are required to carry the comparatively low-pressure steam. Fig. 3 illustrates the method of steam reheating by live steam reheater and indicates clearly the simplicity of such an installation. Owing to the absence of long low-pressure piping, the steam-heated reheater can handle lower pressures than the boiler-room reheater, so that the reheating can be done at a later stage of the expansion.

Since it is the latent heat of the live steam that is doing most of the heating, it is only necessary to have a small quantity of live steam. Therefore, a small pipe is all that is required, and this may be taken off the main steam line leading to the turbine throttle. The temperature to which low-pressure steam can be heated by high-pressure superheated steam is approximately 30 deg. F. above the saturation temperature of the high-pressure steam. The additional heat is obtained from the superheat of the live steam.

It is true that the low-pressure steam cannot be heated to the total temperature of the live superheated

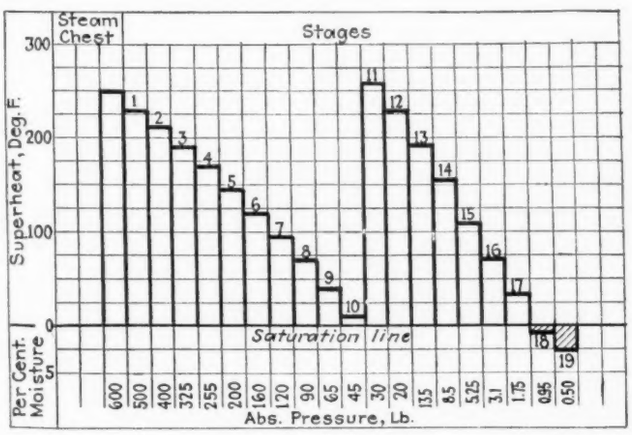


Fig. 6—Steam reheating at lower pressure accomplishes about the same results as shown in Fig. 5

steam. The thermodynamic gain is, therefore, not so great as when the steam is reheated in a boiler to the final temperature of the live steam. It is possible, however, by reheating the steam by this method to keep practically all the stages of the turbine dry, the same as with highly reheated steam, thereby gaining the full practical advantage of reheating. With this method the steam can be reheated at a lower pressure than by any of the other reheating methods, since, because no long piping is required, the live steam reheater is capable of handling much larger volumes of steam than is possible with a reheater located in the boiler setting.

Figs. 4, 5 and 6 show diagrammatically the condition of the steam in the various stages of a nineteen-stage turbine without reheating and with reheating at two different temperatures. Fig. 4 shows the large number of stages of a turbine in which moisture is found and the considerable percentage of moisture in the last stage when steam of 600 lb. pressure is used without reheating. The cross-sectioned area represents the stages in which the steam is wet. Fig. 5 shows the condition of the steam in the various stages of a turbine in which the steam is taken off at the seventh stage and returned to the eighth after having been reheated to a temperature of 700 deg. Even with this high reheat there is still a small amount of moisture in the last two stages, as shown by the cross-sectioned area. Fig. 6 shows the steam condition for a turbine in which the steam is taken off at the tenth stage and returned to the eleventh, having been reheated in a live steam reheater to a temperature of 515 deg., which is 30 deg. above the saturation temperature of the live steam, or 280 deg. F. superheat. Even with the lower reheat temperature of this case as compared with the preceding, the moisture in the last two stages is the same, due to the lower reheating pressure.

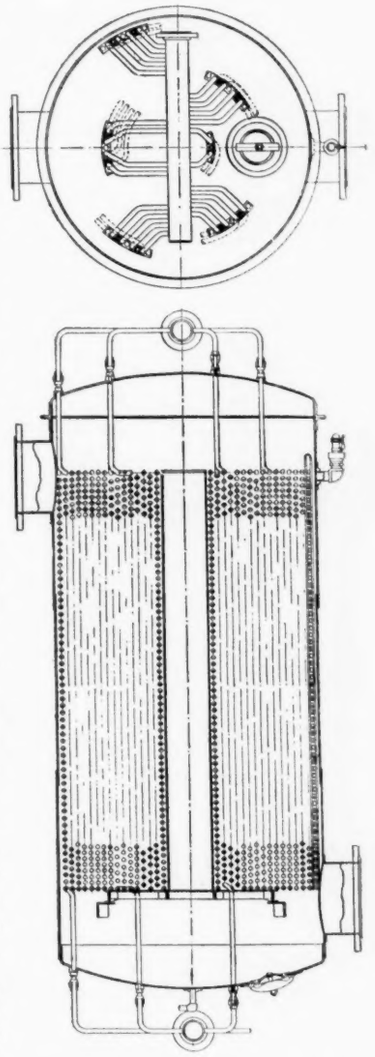


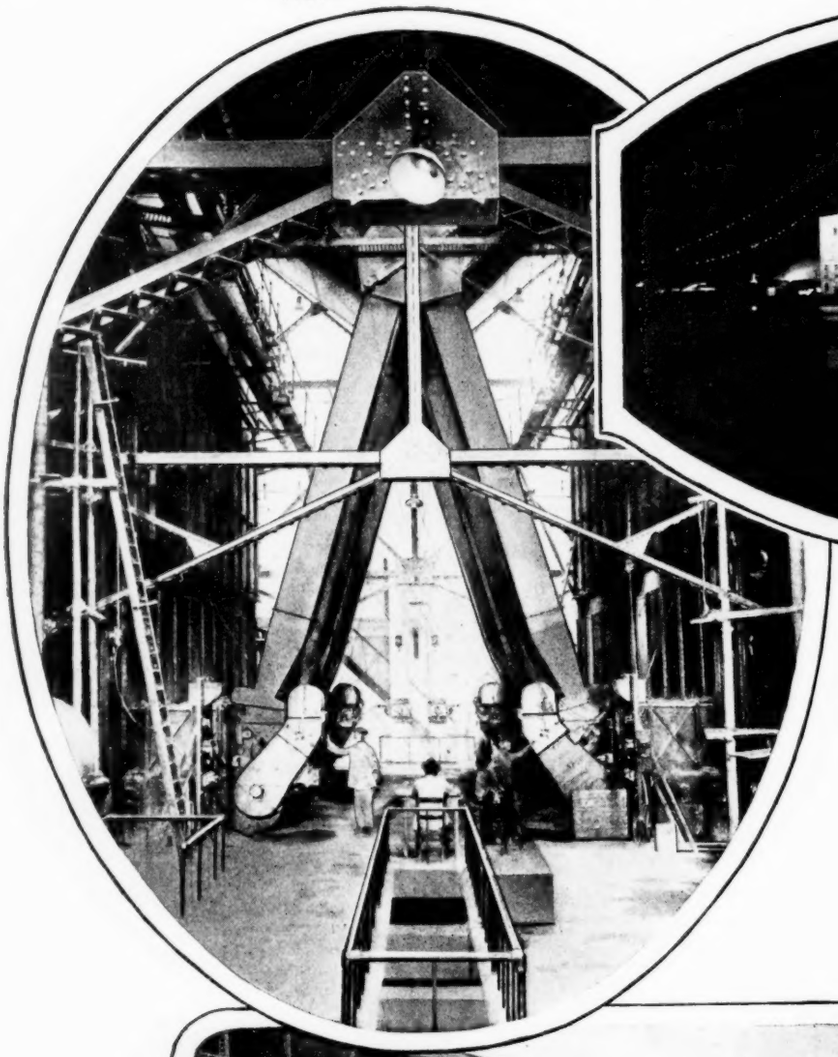
Fig. 7—A design for a steam-heated reheater

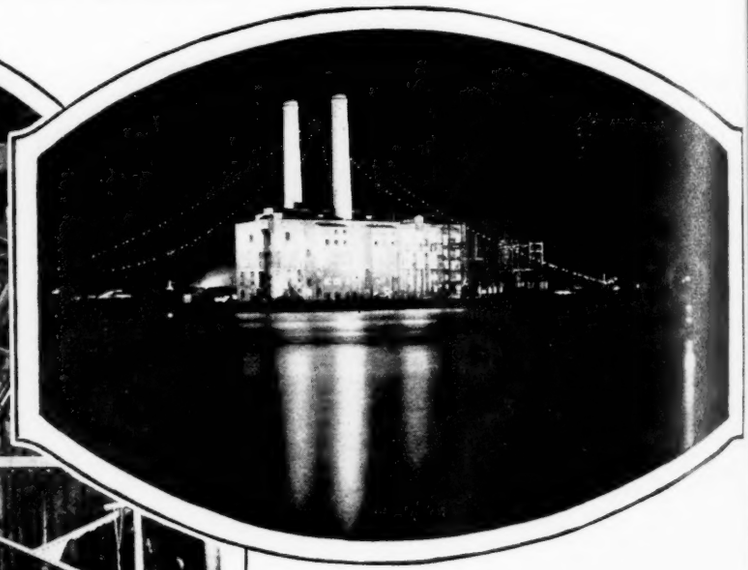
It is clear from these diagrams that the main object of reheating—that is, having dry steam in all the stages possible—is as fully obtained with the live steam reheater as with other types.

A sectional view of a steam reheater designed for 400,000 lb. of steam per hour is shown by Fig. 7. Steam to be reheated flows outside the coils carrying the high-pressure heating steam. Counter-current flow makes effective use of the superheat of the high-pressure steam.

The low pressure drop through the live steam reheater more than offsets the slightly lower thermodynamic efficiency.

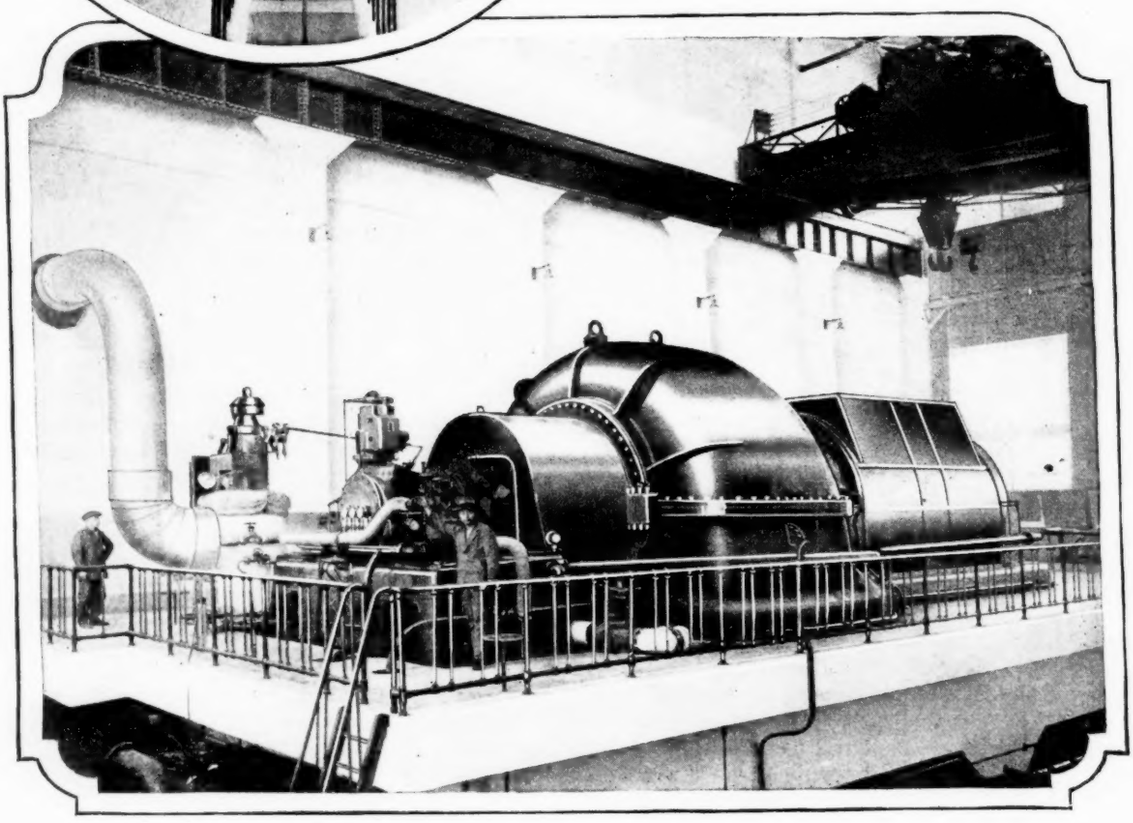
The simplicity of this method of reheating makes it especially desirable for modern power plants already burdened by complicated apparatus and piping.

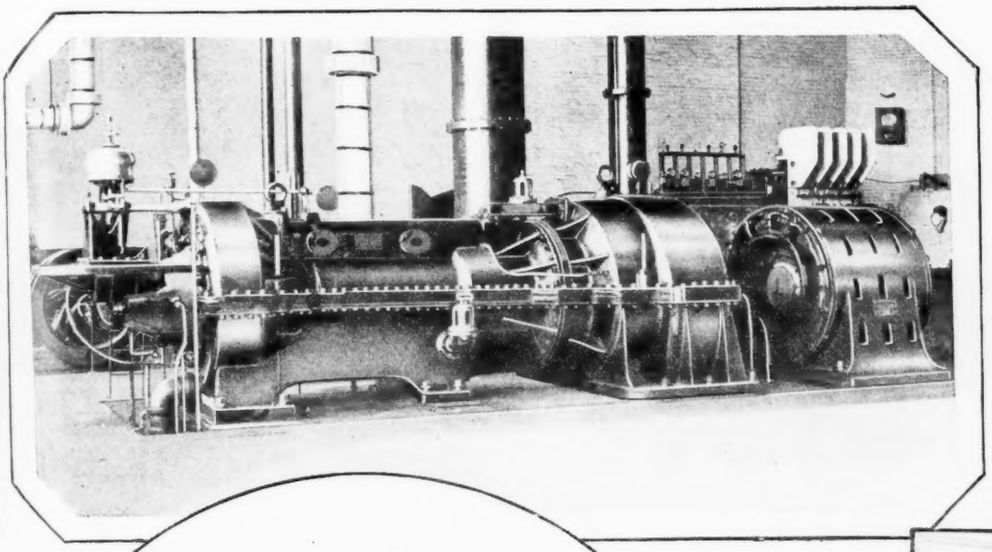




Japan Builds Modern Plant

The new steam-electric station erected in the City of Nagoya, Japan, for the Toho Electric Power Co. was recently dedicated. One 35,000-kw. G.E. turbine (below) is now in operation, and a duplicate will be running before the end of the year. The ultimate capacity will be 140,000 kw. Four water-tube boilers (left), fired by three chain-grate stokers each, supply throttle steam at 315 lb., 735 deg. F. Most of the equipment was imported from America and England, although some was built in Japan.



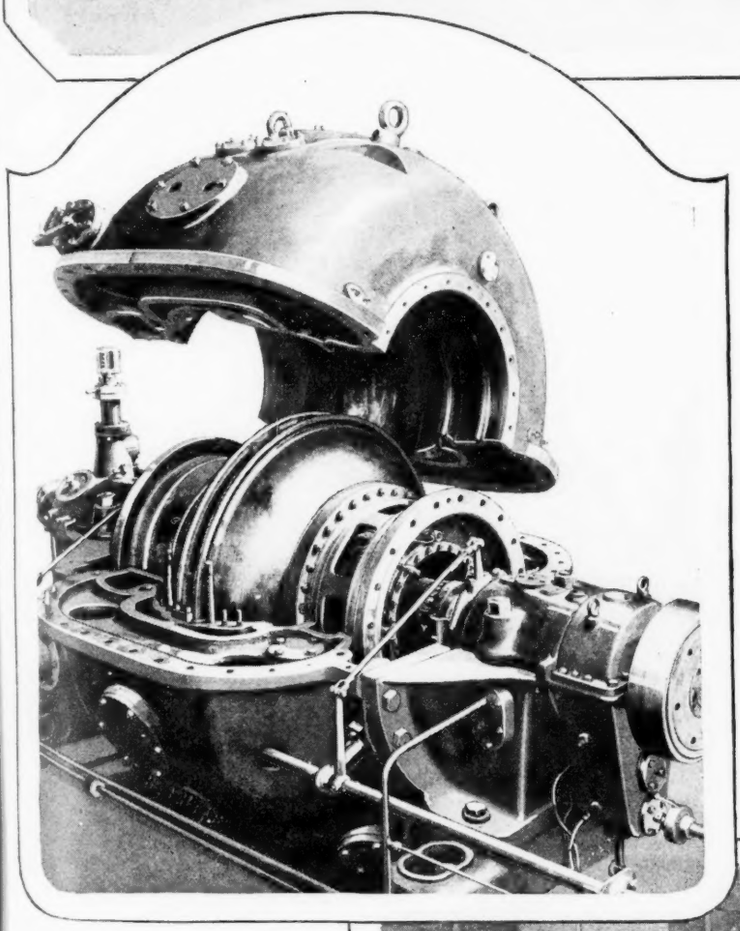


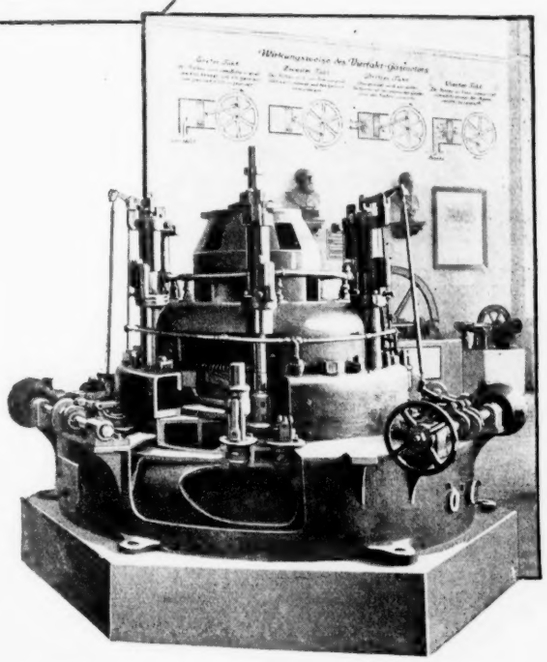
Two historic turbines

Left—Installed in Hartford, Conn., in 1900, this 2,000-kw. Westinghouse turbine was the first horizontal machine in commercial operation in the United States

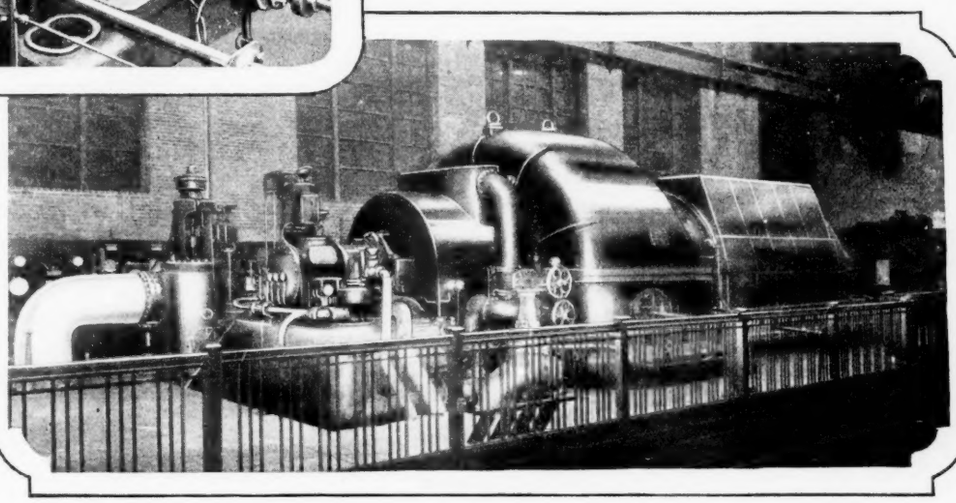
Below—Original Holzworth gas turbine on exhibition in the German Museum at Munich. This, the first gas turbine ever operated, was built in 1908 and developed 30 hp. at 3,000 r.p.m.

Photo by Courtesy of "V. D. I."





Below—This new 35,000-kw. unit in the South St. Station, Providence, bleeds up to 60 per cent of the total throttle steam.



Above — Prof. Stauber's gas turbine, the latest invention in this field, is attracting much attention in Germany. The impeller is operated by water set in motion by gas explosions, as in the Humphrey pump.

Photo by Courtesy of "V. D. I."

Choosing Economical Thickness of Heat Insulation

THE charts in this article supply a rational and easily workable basis for choosing the most economical thickness of heat insulation for any given set of conditions.

By L. E. WHITAKER

WITH fuel costs at their present level and with little likelihood of their being lowered, it will be found that plant superintendents and engineers cannot well afford to neglect the problem of proper heat insulation. The word "proper" has been used advisedly, for, while most engineers realize in a general way the need of covering on certain types of equipment, comparatively few have made a study of this problem, which, with the higher operating temperatures now being used, is becoming one of great importance.

Engineers should not be criticized for lack of interest in, or knowledge of, the problem in the past, for in addition to the factor of cheap fuel, which made

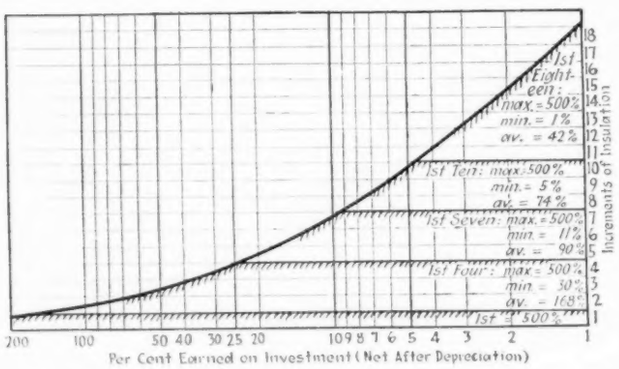

1st Eighteen: max. = 500% min. = 1% av. = 42%
1st Ten: max. = 500% min. = 5% av. = 74%
1st Seven: max. = 500% min. = 11% av. = 90%
1st Four: max. = 500% min. = 30% av. = 168%
1st = 500%
Increments of Insulation
200 100 50 40 30 25 20 10 9 8 7 6 5 4 3 2 1
Per Cent Earned on Investment (Net After Depreciation)

Fig. 1—Curve illustrating fact that the percentage earned on the last increment of thickness is the determining factor

such a study seem unnecessary, there was the fact that very little research work had been done and the experiments carried on by different investigators were incomplete. Such information as was available had been set up in a manner that made fair comparison of values very difficult.

Recommendations made by manufacturers as to type and thickness of insulating materials to be used under different conditions may be right as far as they go, but in every case must be based on assumed conditions. Even though fair average values are chosen, it is more than likely that some of the conditions in any particular case will vary from those assumed. The writer has therefore attempted to arrive at some method that will make it possible for an engineering executive to apply reliable data to his own particular case.

On analyzing the problem, it seemed advisable to divide the controlling factors into two classes: First, there are those—such as heat content of fuel used, specific gravity of fuel oil, number of hours equipment is in use per day and standard units of measure—that do not depend on location of plant, market conditions, etc., and are not subject to a great deal of variation. Then there are others—such as life of installation, cost of fuel, days of operation per year, boiler or still efficiency, and desired return on investment—that will be different in each case considered.

It is obvious that if all these variables are to be taken into account, computations must be made in some detail for each case. However, by assuming values for the first class of variables and for those in the second class that are least subject to fluctuation, it is possible to work out a chart that can be used in connection with heat-loss curves to determine the amount of money one is warranted in spending to secure any given increase in heat saving.

Except in cases where insulation is used to secure only temperature control and where a certain cooling effect is desired, the problem resolves itself into one of balancing first cost against heat loss through the section, to the end that investment in the last increment of thickness results in a net return at least equal to the rate of interest set as a minimum. As the first layer of insulation usually results in a net return of from 200 to 1,000 per cent and each additional thickness results in an additional saving less than that for the last equal thickness, it is apparent that, if figures are based on over-all saving of the entire section, there will be some part of the investment that will not earn the desired interest on investment. It is true that the installation as a whole will soon pay for itself, but, in doing so, the inner layers, whose earning power is great, will have to carry the outer layers, some of which would not be able to earn even depreciation.

The following method is one that appeals to the writer as being a fairly direct way of arriving at the desired condition, in which the capital investment for the extra first cost of the last increment of thickness can be shown to earn the minimum return desired.

First Layer Earns Much More Than Others

The relative earning power of various increments of thickness of insulation is illustrated in Fig. 1. Values have been chosen arbitrarily, so this curve shows only a general trend. A similar curve, plotted accurately and representing an actual set of conditions, would have all the general characteristics of the one shown, in that the net return from the first layer would be very high, that the rate would decrease for each increment and that beyond a certain point earning power would be so small that even depreciation could not be taken care of.

As may be seen, in Fig. 1, each of the first four increments of thickness results in net earnings of over 25 per cent. Additional thicknesses, 5 to 10, each earns more than 5 per cent on the investment, but the heat saving due to further increments is so small that, even when one is satisfied with a low return, the expenditure

necessary to secure them is not warranted. The advisability of recognizing this condition is clearly demonstrated by the fact that, while the average return for the first eighteen increments is 42 per cent, the net return on cost of the last increment of thickness is only 1.1 per cent, which is below any reasonable minimum interest rate.

In view of the foregoing conditions, it is apparent that, in using heat-loss curves and the chart shown in Fig. 2 for arriving at economical thickness, care should be taken to compare any section under consideration with the next commercial thickness. When only such small differences in thickness are considered one is able to approach very closely the ideal condition, in which the last small increment earns just the minimum return decided upon. If larger differences are allowed, it will be found that, while additional expenditures may seem to be warranted, one would actually be using too much insulation. This is true because the figures apply to an average over too large a part of the curve shown in Fig. 1, and the minimum return, which should be the limiting condition, is much lower than the average return for the large increment considered. *Always limit comparisons to the smallest practical difference in thickness.*

The solution of our problem would be simplified if all manufacturers would set up heat-loss figures so that total loss through one linear (or square) foot of a given section could be determined for any temperature difference (pipe to room) by simply referring to a table or chart. This is desirable, because, in the final analysis, it is the total heat loss or saving, not the efficiency of section or loss per degree difference in temperature, that must be compared with the cost of insulation. The foregoing units (square foot and linear foot) have been chosen because they are the ones most commonly used in pricing materials.

In order to simplify the procedure, it was deemed advisable to assume values for all except the two most important variables, namely life of installation and value of fuel fired. The assumptions made in plotting Fig. 2 are given in the caption. If these values are appreciably different from the correct ones, in any case under consideration, it will be an easy matter to prepare a similar chart using data that apply to the case at hand. Of the assumptions made, the following are most subject to variation: number of days in service per year, return on investment and boiler or still efficiency. The other figures will undoubtedly be reasonably close to correct values, in most cases.

The values shown in Fig. 2 were arrived at by equating "annual saving due to reduction in heat loss of 10 B.t.u. per hour" and "annual cost, which includes depreciation and the desired interest on investment." In using only this chart, the resultant figures apply to no particular unit of measurement, for, as the reader will appreciate, the saving of 10 B.t.u. per hour, regardless of the method used, will warrant some certain maximum capital expenditure. However, as heat-loss curves are plotted to show values per square foot or per linear foot, as the case may be, the allowable expenditure, as found from Fig. 2 when used in connection with heat-loss figures, then applies to these same units. As curves representing values for various lengths of life of installation are straight lines and begin at zero of both co-ordinates, Fig. 2 can be plotted by computing a single value for each life of installation considered.

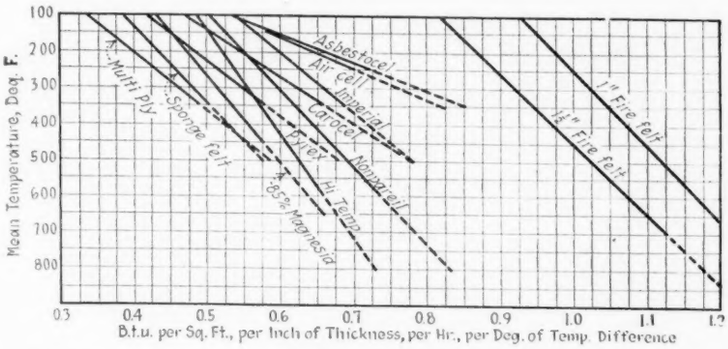


Fig. 3—Absolute conductivity of various insulating materials

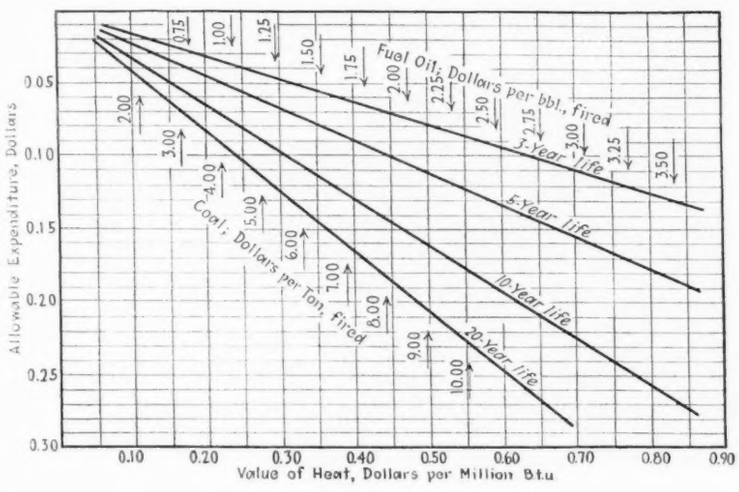
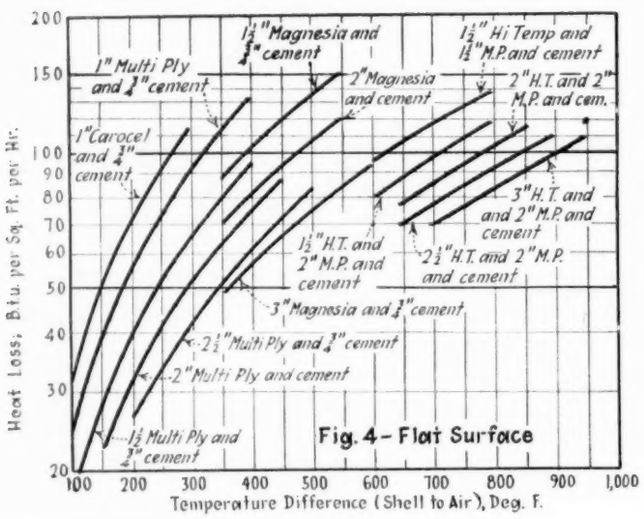
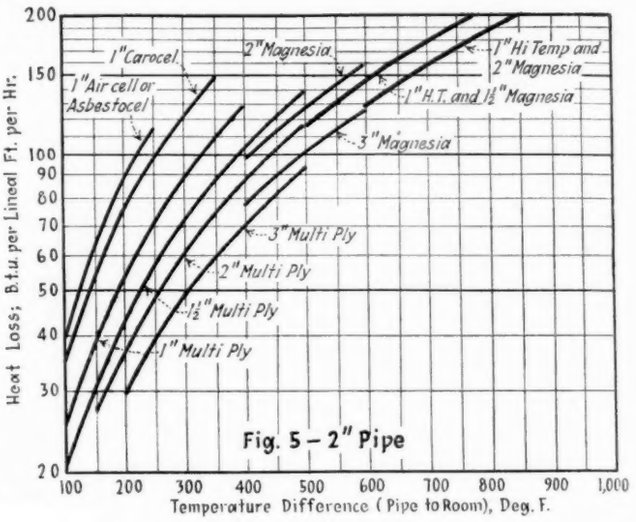


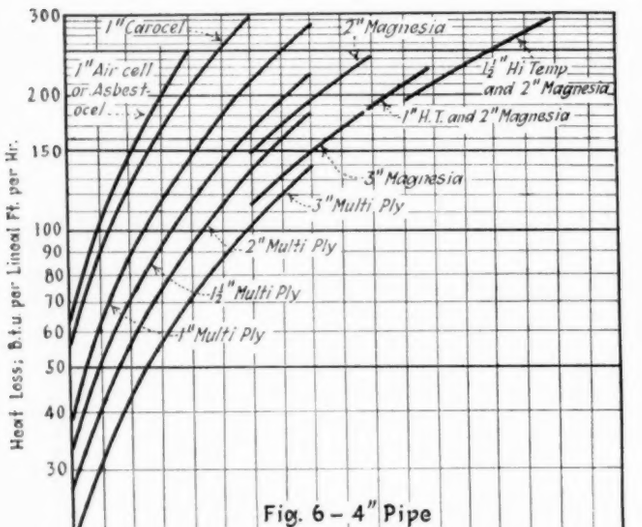
Fig. 2—Chart giving allowable expenditure to save 10 B.t.u. per hour at the insulated surface. It is based on the following assumptions: 300 operating days per year; 24 operating hours per day; 2,000 lb. of coal per ton; 13,000 B.t.u. per lb. of coal; 42 gal. per barrel of oil; 7.75 lb. per gallon of oil; 18,500 B.t.u. per lb. of oil; 70 per cent boiler or still efficiency; 25 per cent minimum return on investment

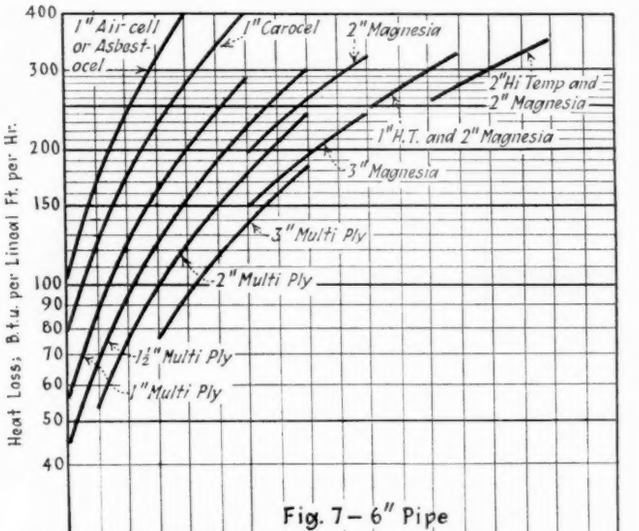
Fig. 3 shows the *absolute* conductivity of various commercial heat-insulating materials as determined by R. H. Heilman.[1] Inasmuch as these values are plotted against mean temperature, they may be used in computing losses through any desired combination of materials. These values have been used in arriving at

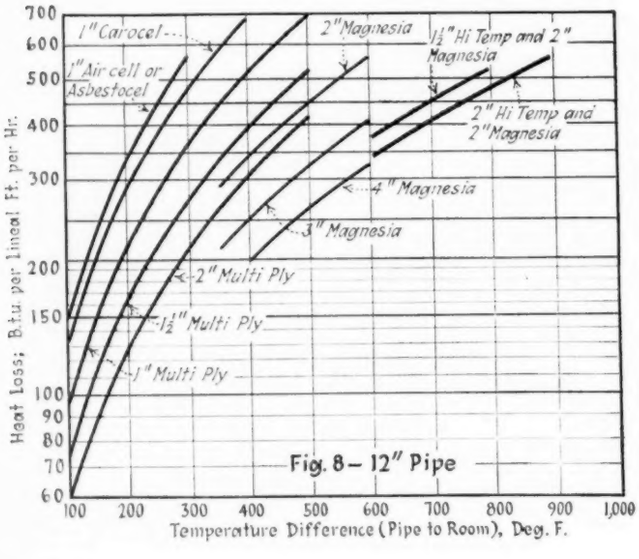
[1] Industrial Fellow, Mellon Institute, University of Pittsburgh.


1½" Magnesia and ¾" cement
1½" Hi Temp and 1½" M.P. and cement
1" Multi Ply and ¾" cement
2" Magnesia and cement
2" H.T. and 2" M.P. and cem.
1" Carocel and ¾" cement
3" H.T. and and 2" M.P. and cement
1½" H.T. and 2" M.P. and cement
2½" H.T. and 2" M.P. and cement
3" Magnesia and ¾" cement
2½" Multi Ply and ¾" cement
2" Multi Ply and cement
1½ Multi Ply and ¾" cement
Fig. 4 – Flat Surface
Heat Loss; B.t.u. per Sq. Ft. per Hr.
Temperature Difference (Shell to Air), Deg. F.


1" Carocel
2" Magnesia
1" Hi Temp and 2" Magnesia
1" Air cell or Asbestocel
1" H.T. and 1½" Magnesia
3" Magnesia
3" Multi Ply
2" Multi Ply
1½" Multi Ply
1" Multi Ply
Fig. 5 – 2" Pipe
Heat Loss; B.t.u. per Lineal Ft. per Hr.
Temperature Difference (Pipe to Room), Deg. F.


1" Carocel
2" Magnesia
1" Air cell or Asbestocel
1½" Hi Temp and 2" Magnesia
1" H.T. and 2" Magnesia
3" Magnesia
3" Multi Ply
2" Multi Ply
1½" Multi Ply
1" Multi Ply
Fig. 6 – 4" Pipe
Heat Loss; B.t.u. per Lineal Ft. per Hr.
Temperature Difference (Pipe to Room), Deg. F.


1" Air cell or Asbestocel
1" Carocel
2" Magnesia
2" Hi Temp and 2" Magnesia
1" H.T. and 2" Magnesia
3" Magnesia
3" Multi Ply
2" Multi Ply
1½" Multi Ply
1" Multi Ply
Fig. 7 – 6" Pipe
Heat Loss; B.t.u. per Lineal Ft. per Hr.
Temperature Difference (Pipe to Room), Deg. F.


1" Carocel
2" Magnesia
1½" Hi Temp and 2" Magnesia
1" Air cell or Asbestocel
2" Hi Temp and 2" Magnesia
4" Magnesia
3" Magnesia
2" Multi Ply
1½" Multi Ply
1" Multi Ply
Fig. 8 – 12" Pipe
Heat Loss; B.t.u. per Lineal Ft. per Hr.
Temperature Difference (Pipe to Room), Deg. F.

Figs. 4 to 8—Heat losses through various combinations of insulating materials on flat surfaces and on pipes of various diameters

These are used in connection with Fig. 2 and data on the given conditions to choose the most economical combination of materials and thickness. This changes but slowly with the curvature of the surface, so the insulation found suitable for one pipe size will also be suitable for pipes slightly larger or smaller. The economical combination for a flat surface will also be suitable for a boiler drum or other cylindrical surface of gradual curvature.

heat losses (Figs. 4 to 8), because, in addition to the fact that they are set up in usable form, they are the result of an extensive and accurate series of tests carried on by a research organization of national standing. In the writer's opinion, Mr. Heilman's paper, presented before the American Society of Mechanical Engineers at the May, 1924, meeting, is the most complete and most valuable publication on heat insulation available at this time.

The universally accepted formulas[2] for computing heat losses through insulating materials have been used in arriving at values given in Figs. 4 to 8. These formulas have been verified by so many tests that there is no question as to their being correct, at least within practical limits. Even though they were slightly in error, their having been used in arriving at absolute conductivities would tend to compensate for the error.

Assumptions Made in Plotting Curves

The actual method of computing heat losses used is that outlined by Mr. Heilman in the paper just mentioned. It has been assumed that the temperature of the metal surface and the inside surface of the insulation will be the same and that, where two layers of material are used, the outside-surface temperature of the inner layer and the inside-surface temperature of the outer layer will be the same. Any error introduced by these assumptions is too small to make any practical difference, and inasmuch as they involve neglecting small temperature drops in the section, they result in figures that are conservative rather than over-optimistic.

As space will not permit the plotting of curves for all types or kinds of coverings that might be under consideration, it was deemed advisable to show a fairly complete set of curves for Multi-Ply (a laminated asbestos insulation) with the suggestion that in making comparisons, the table of factors given in the accompanying table be used. These are taken from Table V in Mr. Heilman's paper. The product of the proper factor and the heat loss through Multi-Ply covering gives the heat loss through the covering under consideration.

PIPE-COVERING FACTORS

Type of Covering	Temperature Difference, Pipe to Air—100° F.	200° F.	300° F.	400° F.	500° F.	600° F.	700° F.
85% magnesia	1.232	1.185	1.151	1.116	1.090	1.062	1.040
Nonpareil	1.415	1.361	1.313	1.279	1.251	1.220	1.196
Asbestos-Sponge Felted	1.045	1.038	1.033	1.026	1.021	1.014	1.010
Hi-Temp	1.347	1.285	1.230	1.185	1.150	1.115	1.085
Imperial	1.515	1.474	1.441	1.409	1.383	1.360	1.345
Carocel	1.349	1.340	1.333	1.323	.....	.....	.....
Pyrex	1.200	1.196	1.186	1.175	1.172	1.166	1.160
Air Cell	1.520	1.550	1.580	1.605	.....	.....	.....
Asbestocel	1.535	1.565	1.610	1.649	.....	.....	.....
Fire-Felt, 1 in. thick	2.206	2.118	2.040	1.979	1.927	1.877	1.840
Fire-felt, 1.5 in. thick	2.090	2.011	1.927	1.865	1.811	1.761	1.722

In using these factors to obtain heat loss through covering other than Multi-Ply, it should be borne in mind that results will be reasonably close approximations—not accurate values.

If values for heat loss through different thicknesses of covering were desired for each pipe size, it would be necessary to have a large number of tables or curves. For the purpose of arriving at economical sections of insulation this will not be necessary, for so far as economical thicknesses are concerned, a thickness for one pipe size will apply to others that are reasonably close to the one for which detailed values are worked out.

An example showing the suggested use of charts will probably make the method clearer than a lengthy discussion. The following conditions will be assumed as those prevailing in some particular case where there is doubt as to the thickness of insulation that should be used.

Superheated steam temperature, deg. F	750
Air temperature, deg. F	70
Fuel value dollars per bbl. of oil (fired)	$1.50
Following costs for insulation applied, dollars per sq.ft.	
3-in. block thickness plus ¾-in. cement	$1.00
3½-in. block thickness plus ¾-in. cement	1.10
4-in. block thickness plus ¾-in. cement	1.25
4½-in. block thickness plus ¾-in. cement	1.40
Minimum return on investment, per cent	25
Useful life of installation, years	10

Inasmuch as the temperature encountered is higher than that at which 85 per cent magnesia or laminated asbestos begins to fail (the usual accepted limits are 600 deg. F. for magnesia and 400 deg. for asbestos paper), it will be necessary to use some form of high-temperature insulation for at least the first layer. As a general rule it will be found that, if approximately one-half the total block thickness is composed of the high-temperature insulation, the maximum temperature in the outer layer of the blocks will be within safe limits for the more effective insulators.

Some materials capable of withstanding temperatures up to 1,000 deg. F. are good insulators, but as none of them is equal, in the lower temperature range, to 85 per cent magnesia or the higher-grade laminated asbestos coverings, it will in most cases, be advisable to use a combined section composed of an inner layer of high-temperature insulation and an outer layer of the more effective materials.

Practical Problems Worked Out

As the manufacturers' recommendations are probably based on economical sections for average conditions, it will be well to compare the heat loss through the installation recommended for the assumed temperature difference (750 — 70 = 680 deg.) with the heat loss through sections ½ in. thinner and ½ in. thicker.

The section recommended by one manufacturer for this temperature is 2-in.-thick Hi-Temp, 2-in.-thick Multi-Ply (or 85 per cent magnesia) and ¾-in.-thick Asbestos-cement plaster coat. The loss through this section, when Multi-Ply is used as the outer layer, is 82.5 B.t.u. per sq.ft. per hour (see Fig. 4, showing losses through insulation on flat or large-radius curved surfaces). For the same temperature difference it will be apparent from Fig. 4 that the loss through 1½ in. Hi-Temp, 2-in. Multi-Ply and ¾-in. cement is 93.5 B.t.u. per sq.ft. per hour and that the corresponding loss for 2½-in. Hi-Temp, 2-in. Multi-Ply and ¾-in. cement is 74.5 B.t.u. per sq.ft. per hour.

By referring to Fig. 2, it will be found that, under assumed conditions, a saving of 10 B.t.u. per hour will be worth while if it does not require an additional investment in excess of $0.115. The recommended section saves 11 B.t.u. per sq.ft. per hour more than the next thinner one. The product of $0.115 and 1.1 will then give the allowable expenditure, which is $0.127 per sq.ft. This is so close to the $0.15 difference in cost that either section could be considered as being satisfactory. If conditions are as assumed in Fig. 2, and 25 per cent net return is an absolute minimum, it is probable that the less expensive section will be chosen, but if there are reasons for believing that an average of fuel costs over the life of installation will be greater than the $0.355 per million B.t.u. assumed, it would seem advisable to use the thicker section.

[2] See A. S. M. E. *Proceedings*, May, 1922.

In case it is decided to use a total block thickness of only 3 in., it will be advisable to use 1½ in. of Hi-Temp and 1½ in. of Multi-Ply in order to make sure that the inside surface temperature for the outer layer will be reduced to within safe working limits.

The saving in fuel through insulation is usually considered to be the only benefit derived through the use of coverings, but while it is probably the largest and most easily demonstrated gain, there are other advantages that should not be overlooked. One is that a well-insulated system will secure the same results as a poorly insulated one, with less forcing. This results not only in reduced fuel consumption, but in greater life of equipment and lower maintenance costs.

Where steam must be carried through long lines, it is often necessary to use more covering than the economical thickness called for by fuel value alone.

Each individual heat-insulation problem is worthy of considerable thought, and in the case of large installations the engineering executive in charge of design will do well to go into the matter of type and thickness of material quite thoroughly. In such a case it would be better to compute annual costs and savings accurately, using actual values applying in the particular case at hand for all factors entering into the problem. Then some such method as that outlined may be used as a check and will be found of value in determining thicknesses within which to limit detailed investigation.

The Instrument Layout from the Investment Viewpoint

The Economic Principles Underlying the Intelligent Selection of Power-Plant Instruments

INSTRUMENTS, like other items of power-plant equipment, are sometimes purchased on the basis of hunch, whim or passing fancy. But there is a growing tendency in modern industry to replace prejudice by analysis. It is assumed that the reader of this article approaches the problem of instrument layout as an engineer and a business man—that his sole idea is to invest money wisely.

Before he can make a wise selection of instruments, he must gather data on first cost, rate of depreciation, cost of maintenance and supervision, and the probable gross savings obtainable with each type considered. The last-named item is of primary importance, but is generally hard to estimate because of the intangible nature of some of the elements involved. In most cases only a rough evaluation of these elements is possible, but this is no sound reason for overlooking them. They exist and it is certainly better engineering to make even an "educated guess" at their value than to overlook them entirely.

Most of the present article will be devoted to this question of estimating the gross saving, with an outline of the general method of determining which of several propositions is the best investment.

The most common motive for the purchase of power-plant instruments is an anticipated saving in fuel due to more intelligent operation. In evaluating a fuel saving, the cost of firing should be added to that of the coal as delivered at the plant. Moreover, it sometimes happens that the increased efficiency or capacity made possible by the use of instruments postpones the necessity of enlarging the plant. In such cases the corresponding saving in overhead should be credited to the instruments.

Among the less tangible gains are improved working conditions, the supplying of reliable data for the general accounting system of the manufacturing plant or public utility of which the power plant is a part, reduced outages and generally improved character of the service. The plant with instruments makes more appeal to the intelligent employee. Morale is improved where definite comparative records replace guess and gossip.

The average business spends from one to three per cent of its gross revenue in maintaining the accounting system. Many of the instrument records can enter into these general accounts and thus assist greatly in the intelligent conduct of the business. This is in addition to their use in connection with the power-plant record system. In certain cases instruments have been installed for the sole purpose of obtaining trustworthy information needed as a basis for making plant extensions.

All these gains must be estimated as part of the gross saving due to instruments, although the fuel saving is generally the only one that can be expressed in dollars with anything approaching accuracy.

It will now be worth while to consider some of the applications of instruments in the power plant, noting the particular ways in which they may lead to savings, and outlining methods of estimating some of these savings. To limit the discussion and to avoid going into a field already well standardized, no attention will be paid to switchboard or other strictly electrical instruments.

Starting in the boiler room as the most fertile ground for saving money by instruments, one thinks first of draft gages. There is probably no other equally small power-plant investment that can yield such a large return. Draft gages fall in the class of bread-and-butter essentials. Without their aid intelligent operation of a boiler is impossible. The draft drop through the tubes is a measure of the gas flow and roughly of the boiler output. When the draft through the fire varies appreciably from the normal for the given rating, the intelligent fireman at once looks for a defective fire—too thick or too thin, clogged or full of holes.

With forced draft the damper can be regulated to maintain a small constant draft over the fire. This will greatly reduce the heavy loss due to air infiltration and at the same time protect the refractories from excessive temperatures. The savings obtainable in these various ways are so uniformly large that one may safely install the best draft gages obtainable without going to the trouble of making any computation of the savings. Draft gages, like steam gages and safety valves, should

be installed without question on every power-plant boiler.

The situation with respect to the measurement of feed temperature is much the same. The fact that each 10 deg. rise in feed temperature saves about one per cent of the coal bill is one reason for this. Another is that variations of feed temperature set up injurious strains in the boiler. In most plants the additional control made possible by the recording type of instrument will well repay the extra cost.

Measurements of flue-gas temperature are essential to economical operation. Since it is not in accordance with human nature for firemen to climb ladders every few minutes to inspect mercury thermometers, the flue temperature is almost invariably registered in modern plants by instruments on the operating level. Here again the recording instrument permits a check-up. It also shows nicely the progressive accumulation of soot and scale.

Boiler Instruments Save Much Money

Where it is due solely to inefficient heat absorption in the boiler, the loss from excessive flue temperature amounts to about 5 per cent of the fuel bill for every hundred degrees. The correct flue temperatures for any rating can be determined by operating the boiler in a careful manner just after the unit has been overhauled, tubes cleaned inside and out, baffles repaired and setting made tight. If firing methods continue good, a rising flue temperature indicates scaled or soot-covered tubes, or leaky baffles. Without an instrument the fireman would be unlikely to pick the correct time for cleaning and repairs.

Rises in flue temperature are also caused by excess air due to leaky setting, holes in the fire, etc. In this case the loss is much more than 5 per cent of the fuel bill because the higher temperature is merely a symptom of the greater quantities of heated gases passing to waste.

The two most important measurements in a power plant are the coal consumed and the electrical energy generated. These two tell the story of the over-all performance of the plant. Combined with the steam output, the coal weight likewise gives the boiler-room performance. In addition to providing for weighing the coal as delivered to the bunkers, it will generally pay to weigh, by larry or otherwise, the coal supplied to individual boilers. In connection with steam meters or feed-water meters, this permits determination of the over-all performance of individual boilers. It shows which boilers are at fault when the performance of the boiler room as a whole falls off.

Feed-water meters, steam-flow meters (on boilers), CO_2 recorders and steam-flow-air-flow meters should be considered in connection with each other because the use of one affects the application of the others to some extent. It is almost universally conceded that there should be some method of measuring the total evaporation, since this represents the boiler-room output. With very small boilers a single measurement for the boiler plant is sometimes considered sufficient, but with boilers of medium or large size individual measurement is generally favored, with group or boiler-room measurement as a check. Experience shows that a continuous check on the evaporation per pound of coal will reduce fuel consumption at least 5 per cent in the average plant and sometimes much more, due to the continuous incentive it gives the operatives to make a good record.

The only question that ordinarily arises in this connection is whether a feed-water meter, a steam meter or both should be used. Considered individually, each has its advantages. The best water meters are generally somewhat more accurate than steam meters, because of the unavoidable variations in steam pressure and quality (or superheat). On the other hand, a good steam meter is reasonably accurate and has certain advantages. The water meter requires deduction of the blowdown before the evaporation can be obtained. Only the steam meter can give a true instantaneous reading of the rate of evaporation, since the rate of feed never keeps exactly in step with the evaporation except in certain rare cases where the load is absolutely constant.

To get all the advantages of both types of meters, the larger and better-equipped boiler plants generally use a combination of feed-water meters on groups of boilers and steam meters on the individual boilers. Some of the smaller plants, however, feel limited to the choice of one or the other.

The decision is further complicated by the primary importance of maintaining the proper ratio of air to fuel burned to keep down the chimney losses. Where no instruments are provided to guide the firemen in maintaining this ratio, the resulting unnecessary waste of fuel often amounts to 10 per cent of the total fuel consumed.

One common method of watching the fuel-air ratio is to measure the CO_2 in the flue gas either automatically or by analyzing occasional snap samples in an Orsat. The other is to use a steam-flow-air-flow meter, which helps the fireman to maintain a constant ratio between steam production and the flow of gas through the boiler. If he succeeds in this, the air-fuel ratio is automatically maintained. The complication arises from the fact that the selection of a steam-flow-air-flow meter obviously equips the boiler with a steam-flow meter and to that extent reduces the necessity for a feed-water meter. The previously mentioned compromise, slightly modified to give steam-flow-air-flow meters on each boiler and a feed-water meter for each group of boilers, would fit in nicely here.

Where steam-flow-air-flow meters are installed, the automatic CO_2 recorder is sometimes omitted, a hand Orsat being used for occasional check up of the settings of the instrument pens for the proper ratio of fuel to gases.

Recorders for Combustible Gases

In the matter of keeping down flue-gas losses there is one limitation which the ordinary CO_2 recorder shares with the steam-flow-air-flow meter. That is the failure to indicate the loss due to incomplete combustion. A combination recorder for CO_2 and combustible gases permits operating at high CO_2 without stepping over the danger line of incomplete combustion. Where combustion is controlled by steam-flow-air-flow meters or by a CO_2 recorder, the flue gases should be occasionally checked for combustibles with a hand Orsat.

Returning to steam meters, their field extends far beyond the boiler room. Wherever large quantities of steam are delivered to any steam-consuming unit or group of such units, the advisability of applying such a meter should be considered. The most obvious location outside of the boiler room is in the line to each main unit. In connection with the watt-hour meter the steam meter permits instant detection of any increase

in steam rate. If vacuum and steam conditions are normal, erosion or other trouble in the turbine is indicated.

In the same way a steam meter on the line to a group of auxiliaries will at once reveal any general falling off of auxiliary efficiency due to pump slip, leaky steam valves, worn impellers, etc.

One of the greatest fields for steam meters is on lines leading to the various processes or departments of a manufacturing establishment. It is human nature to waste that for which one is not held accountable. Experience shows a surprising falling off in steam consumption per unit of product output where the steam to each department is metered and charged against the cost of operating that department.

Other Money-Saving Applications

While recording steam gages and thermometers are widely used to determine the pressure and temperature of the steam leaving the boiler or entering the main unit, it is rather hard to compute the savings in such cases. There is, however, a real saving in fuel in keeping the steam conditions constantly close to the maximum for which the plant was designed. Much is spent to build apparatus for high temperatures and pressures. Failure to take full advantage of the possibilities thus afforded is a waste of money. Incidentally, these records help insure safe and continuous operation.

In addition to the familiar simple mercury-column vacuum gages and barometers used to check condenser operation, there have recently come into use recording instruments to measure these same quantities. These supplement rather than replace the customary gages and are justified in many cases by the extreme importance of maintaining high vacuum.

In large plants instruments are used at many points aside from those mentioned. Some of the modern central stations provide for numerous recording thermometers and pressure gages in the heat-balance system. Feed temperatures are measured at each step from the hotwell through the various feed heaters to the boilers. Steam pressures are measured at each bleed point. Some of these instruments can save but a small fraction of one per cent of the fuel bill, yet enough to justify the cost where the plant is large.

Instruments have recently been developed to bring to light such hidden wastes as excessive blow down or leakage through the blowoff valve, feed-heater overflow, excessive venting, etc. Wherever leaks may be serious and at the same time unobserved, such instruments may easily pay their way.

Other instruments have little connection with fuel consumption, but are justified on the basis of safety, continuous operation and protection of equipment—all matters of considerable financial importance. Among these may be mentioned thermometers for bearing temperatures, and for temperatures of generator cooling air and generator windings.

Some hints have been given as to methods of estimating the gross saving obtainable from the installation of any particular instrument. It is clear that the net return is this gross saving minus depreciation and the cost of supervising and maintaining the instrument.

The problem remains of determining which of various instruments proposed for a given job is the best investment. At the start the principle may be laid down that there should be established a certain minimum percentage that the instrument must return in order to be considered at all. This is generally taken much higher than in commercial security investments because of other high-return opportunities for investing money in the business. Twenty per cent may be taken as a typical figure.

While this percentage return may properly be used as a test to throw out any proposition that falls below the investment "dead line," it cannot, strange to say, be used in comparing two different propositions. It is a common mistake to assume that because one piece of equipment will yield 30 per cent net and another 40, the second is necessarily the better investment. This conclusion follows logically only in the case where the amount invested is the same in both cases.

Investments in apparatus for saving leaks have the peculiar characteristic that, while the percentage return is generally much higher than with security investments, the amount of money that may be invested at a given rate of return is strictly limited. Here an opportunity for enlarging the investment may in some cases, more than compensate for a somewhat lower rate of interest on the total investment. This is nicely illustrated by the problem of the economic thickness of pipe covering as explained by Whitaker in the article on page 762 of this issue.

Two tests should be applied. First, no proposition should be considered which will not yield a net percentage return exceeding the minimum "dead line" established for the given plant. Second, the more expensive of two propositions is the better investment when and only when the *additional* yield from the *additional* expenditure exceeds the established lower limit of net percentage return.

Economic Problem Worked Out

The application of these principles to the choice of instruments can best be illustrated by a concrete example. Assume that the choice has been narrowed down to four propositions as shown in the following table:

COMPARISON OF INVESTMENTS

Proposition	First Cost Installed	Gross Yearly Saving	Depreciation, Supervision and Maintenance	Net Return — Dollars	Net Return — Per Cent
A	$400	$400	$100	$300	75
B	500	100	50	50	10
C	600	500	100	400	67
D	700	550	100	450	64

The last column shows that proposition *B* yields only 10 per cent net and should therefore be discarded on the assumption that 20 per cent is the smallest acceptable net return. The question now arises as to whether it is better to invest $400, $600 or $700. Imagine $700 divided into three parcels, of $400, $200 and $100 respectively. The first $400 can be invested in *A* to yield $300, or 75 per cent, which is entirely satisfactory. Considering item *C*, it is seen that an *additional* $200 investment would earn an *additional* $100 return or 50 per cent. Therefore investment *C* does all that *A* does for the first $400 and permits in addition another worthwhile investment. Obviously *C* is a better investment than *A* although the percentage net return is smaller.

Next compare *C* and *D*. Here a further investment of $100 gives a further yield of $50 or 50 per cent on the additional investment. Therefore *D* is a better investment than *C* and the best in the group, although, leaving *B* out of consideration, it yields the smallest net percentage return on the total investment.

How to Lay Out Power-Plant Piping—I

By SABIN CROCKER
Designing Engineer, Drafting and Surveying Bureau,
The Detroit Edison Company

IN THIS series of articles, the author confines himself to common-sense pointers on how to go about the job of designing a piping layout, with sufficient technical information to indicate the important details which must not be overlooked. This article outlines the preparation of preliminary plans, and later articles will cover the choice of pipe sizes and materials, dimensional standards, allowances for expansion, hanger and anchor location, drips, bypasses and heat insulation.

THE design and construction of piping systems for use under the high pressures and temperatures existing or contemplated in modern power plants involve not only the use of basic principles common to good piping layouts operating under less severe service conditions, but also require the careful working out of a number of features that become extremely important as the service requirements are raised. It is proposed to set forth in this series of articles a brief discussion of several of the major features pertaining to the design and construction of power-plant piping, with special emphasis on points requiring extra attention for severe conditions.

In modern power plants with the extra complications in piping incidental to the use of additional fuel-saving equipment such as economizers, bleeder heaters, reheating boilers, water screens in pulverized coal furnaces, etc., the problem of securing satisfactory piping layouts is far more difficult than in the simpler plants of a few years ago. The multiplicity of pipe lines now required for the interconnection of different apparatus demands careful study to define the function of each line and make its purpose clear in the layout, as well as to insure that it is physically possible to install the piping in a neat and accessible manner without encroaching on space required for equipment, walkways, stairs, light wells or other structural requirements.

On many small jobs it is common practice to set up a pipe bench in the field and cut and fit pipe on the job.

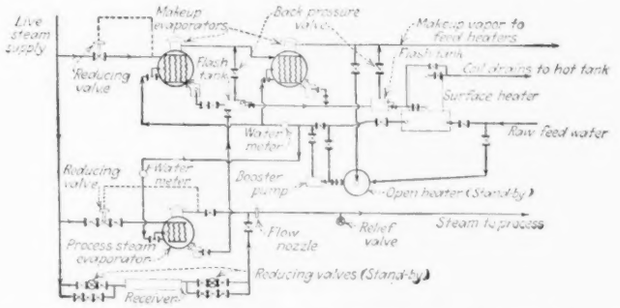


Fig. 1—A preliminary rough sketch shows the layout and gives a chance for study and revision

This seems like a simple and direct procedure, but the result is often a sloppy job at an increased cost. In almost every case it pays well in time, money and general satisfaction to make a detailed layout on the drafting board before starting work.

The use of a line diagram is a time-honored means of explaining the function of each line in a general scheme. The remark is frequently made among engineers that "one cannot talk without a pencil," and this is especially true in explaining a complicated system of piping to the man who is to lay it out on the drafting board. A line diagram serves much the same purpose as the free-hand sketches made during an engineering conversation, but the information conveyed is made available to more people. With involved piping layouts

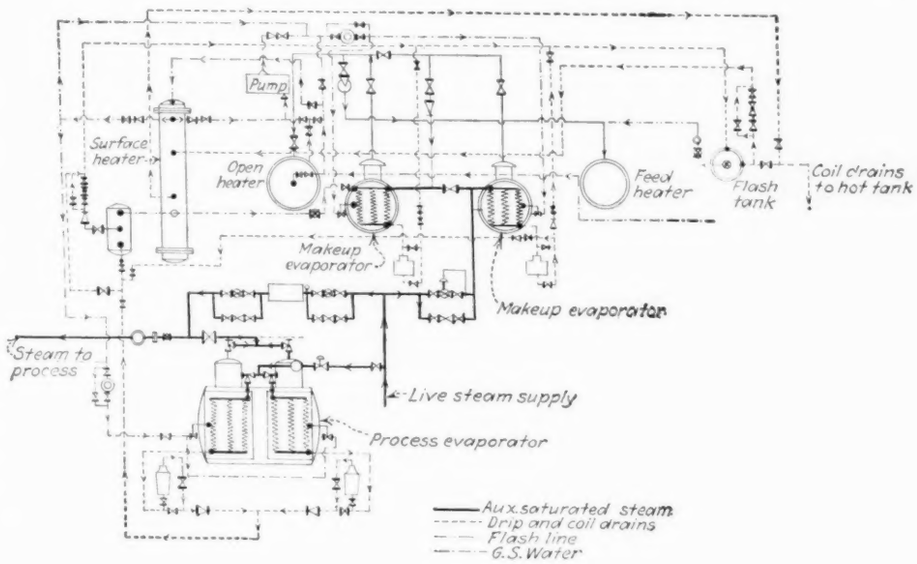


Fig. 2—A more carefully worked out diagram, with all equipment in its actual location, paves the way for space and arrangement studies

In the original, each line was marked to show the nature of the material within it, with pressure, temperature, and rate and direction of flow. Each valve was marked to show its type and use. These detailed markings could not be duplicated in the reduced size of the cut.

it is customary for someone to make a free-hand sketch as a guide for the draftsman, or such a sketch may be made by the draftsman himself and referred to the proper authority for approval before the drawing proper is started. On a large job where considerable engineering study is involved, the engineer who works out the plant heat balance usually makes line diagrams pertaining to heat interchangers in connection with his studies. Such diagrams may either be isometrics, or of the style resembling wiring diagrams. In either case they should show the equipment that is to be interconnected, the function of the different lines, and usually the condition and quantity of the fluid flowing through each. Such a diagram furnishes a good starting point for the piping studies proper.

A diagram of this sort for the combining of two evaporator systems is shown in Fig. 1. In this case a new single-effect evaporator was to be installed in connection with a double-effect evaporator already in use, obtaining its live steam from the same source, but furnishing vapor for an entirely different purpose. Fig. 1 shows the diagram in simple form without regard to the positions of the apparatus. Fig. 2 shows the more complicated diagram which the piping designer worked out from Fig. 1, to take into account the location of existing equipment and piping, and the locations available for the new part of the installation. The involved nature of a job of this kind makes imperative the use of a line diagram, as a guide in laying out the piping, for explanation to the erection crew, and for the intelligent operation of the system.

When a final piping layout is finished and the drawings are nearly ready to send out for bids or to the field erection crew, it is often desirable to reproduce the line diagram in a corner of the layout sheet. Here it is readily available and proves a material aid to ready understanding of the piping system.

After the line diagrams and preliminary piping layouts have been made, it is desirable to make a model

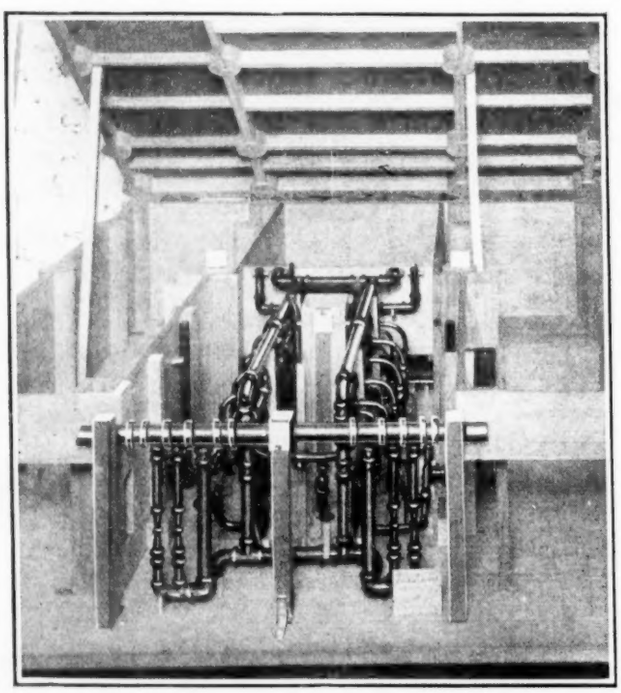


Fig. 3—A well-made model seems an expensive "frill," but really pays for itself in the avoiding of mistakes leading to costly field changes

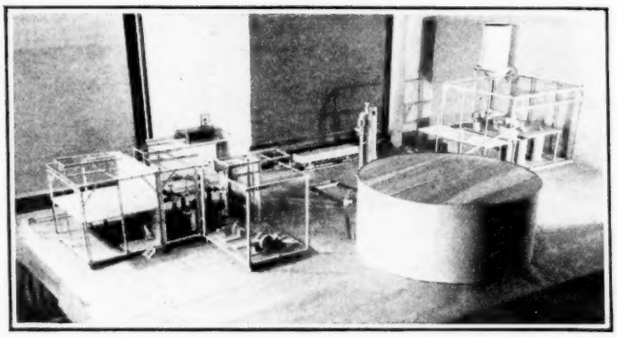


Fig. 4—Where space is not closely limited, the model may be more roughly constructed

of parts of the plant where piping is particularly congested. The use of a model aids in securing a symmetrical layout of pleasing appearance which is conveniently arranged, and at the same time reveals interferences and close clearances. Models may be as extensive or as curtailed as any particular case requires, but they should be made to scale and show the relative positions of piping, equipment and building structure. The proper use of models pays for their cost many times over.

The convenience of a piping layout from an operating point of view is of great importance. The arrangement of a bypass, for instance, should be such that the purpose of each valve is evident at a glance. In cases where there is a repetition of certain units, such as boiler-feed pumps, at intervals throughout the plant, the positions of valves, crossovers, etc., should be as nearly as possible identical. Such duplication is not only desirable from a viewpoint of appearance, but is of immense value to the operating force in the ease with which a given valve can be located in case of emergency. The extra attention given such matters in design is amply repaid by the greater convenience of a well thought out arrangement. It is also frequently desirable to use more pipe and fittings than required for the smallest initial investment, in order to obtain simplicity in operation and the repetition of detail at similar units. A model is of great assistance in working out these features of design, as it reveals at once the advantages in different valve locations and the space left available for access to or dismantling of equipment. Fig. 3 illustrates the result of study of a particularly congested spot. Unfortunately, a photograph cannot do justice to the model in this case.

The amount of detail shown in a model and the accuracy with which it is to scale should depend on the kind of plant which it represents, the congestion of piping in the plant and the desirability of neat appearance. Where there is not sufficient advantage to justify the expense in making a model particularly close to scale, it is possible to make one up out of small pipe and fittings with wood blocks to represent equipment, and structural parts made with children's "Erector" or "Meccanno" sets. A photograph of such a model, for a gas plant, is shown in Fig. 4, reproduced through the courtesy of the Steere Engineering Co. In a power plant the results obtainable through the use of a model justify considerable more expense. The structural parts and equipment should be made closely to scale by a patternmaker. A convenient method of molding the pipe and fittings to scale out of beeswax was described by C. T. VanDusen in an article published in the Feb. 23, 1926, issue of *Power*.

Huge Boiler to Operate at 1,390 Lb. in Lakeside Station

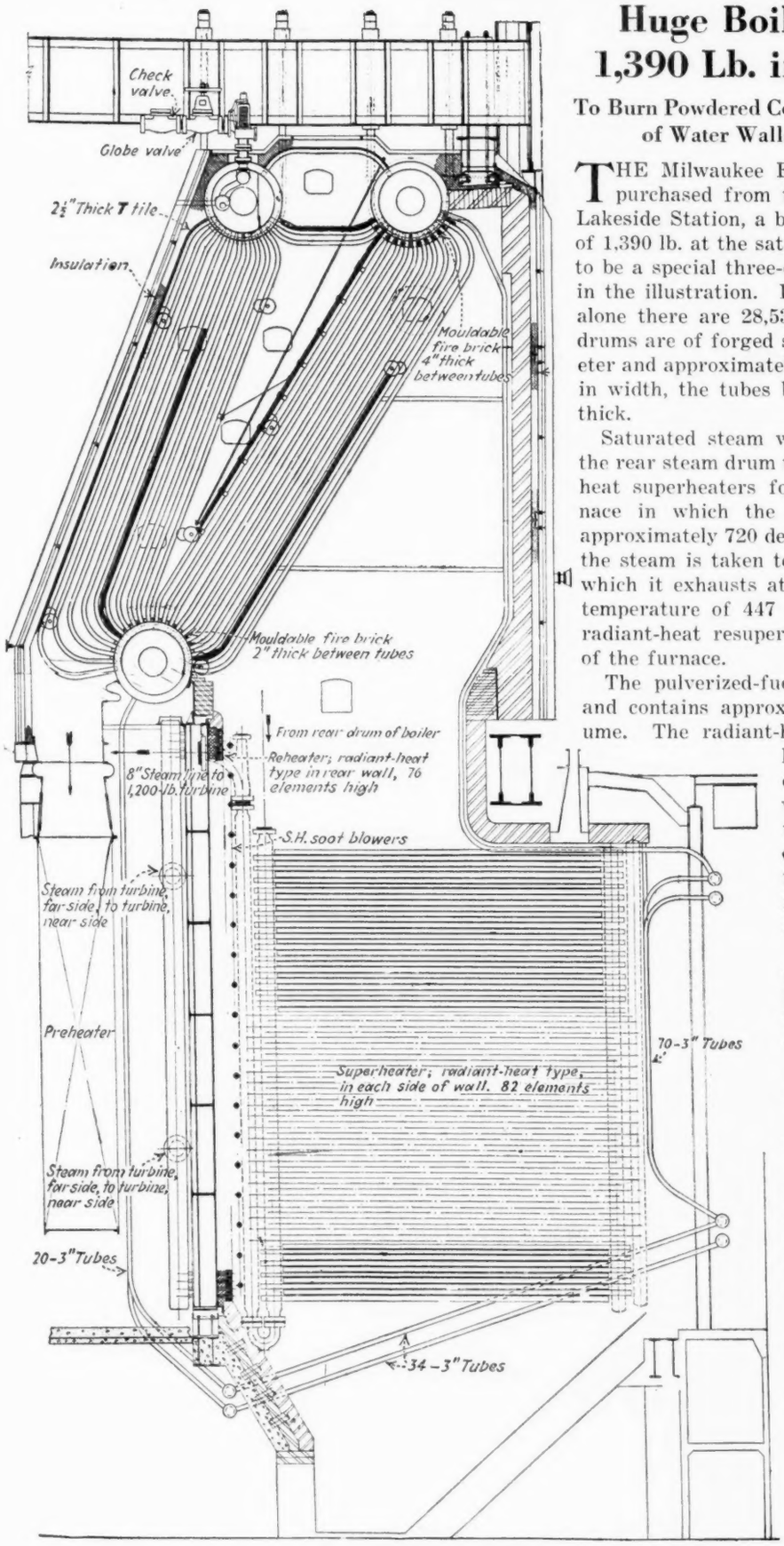
To Burn Powdered Coal and Have Special Arrangement of Water Walls and Radiant Superheaters

THE Milwaukee Electric Railway & Light Co. has purchased from the Babcock & Wilcox Co. for its Lakeside Station, a boiler built for a working pressure of 1,390 lb. at the saturated steam outlet. The boiler is to be a special three-drum Stirling design, as indicated in the illustration. In the pressure parts of the boiler alone there are 28,532 sq.ft. of heating surface. The drums are of forged steel 5 in. thick, 40 in. inside diameter and approximately 40 ft. long. The unit is 70 tubes in width, the tubes being 3 in. in diameter and $\frac{7}{16}$ in. thick.

Saturated steam will be taken from two nozzles on the rear steam drum to two Power Specialty Co. radiant-heat superheaters forming the side walls of the furnace in which the temperature is to be raised to approximately 720 deg. From the primary superheaters the steam is taken to a 1,200-lb.-pressure turbine from which it exhausts at a pressure of 317 lb. gage and a temperature of 447 deg., and is returned to a Foster radiant-heat resuperheater which forms the rear wall of the furnace.

The pulverized-fuel furnace is of the Lopulco type and contains approximately 30,100 ft. of furnace volume. The radiant-heat primary superheater and reheater, as stated, form three sides of the furnace. The front furnace wall, the arch of the furnace and the front boiler wall are water cooled, and a water screen protects the furnace bottom. The front furnace-wall construction is of the Murray fin-tube design.

The gases make three passes over the heating surface and leave the boiler at the bottom of the rear bank of tubes. The flue into which the gases are discharged is divided and the gases are passed downward through two Combustion Engineering Corporation's air heaters, one on either side of the setting. These air heaters are of the plate type, containing approximately 20,160 sq.ft. of heating surface each, and are designed to heat the air to 650 deg. F. when the boiler is operating at maximum load.


Check valve
Globe valve
2½" Thick T tile
Insulation
Mouldable fire brick 4" thick between tubes
Mouldable fire brick 2" thick between tubes
From rear drum of boiler
Reheater; radiant-heat type in rear wall, 76 elements high
8" Steam line to 1,200-lb. turbine
S.H. soot blowers
Steam from turbine, far side, to turbine, near side
Preheater
70-3" Tubes
Superheater; radiant-heat type, in each side of wall, 82 elements high
Steam from turbine, far side, to turbine, near side
20-3" Tubes
34-3" Tubes

Total heating surface in pressure parts of boiler is 28,532 sq.ft. Drums are of forged steel, 5 in. thick, 40 in. inside diameter and 40 ft. long. Radiant superheaters screen the surface side walls and a radiant reheater in the rear wall reheats the steam after expansion from 1,200 lb. to 317 lb.

A news item in this issue describes a trip made last Thursday in New York Harbor by the pioneer German Rotor Ship, "Baden Baden." The inventor, Anton Flettner, is well known for other inventions as well, among them being the Flettner rudder now installed on about 500 airplanes and nearly 100 ships. A 3,000-ton, 3-rotor ship, the "Barbara," was launched in Bremen on April 28. She is equipped with three rotors 55 ft. high and 12 ft. in diameter.

Industry's Boiler Equipment

A Statistical Analysis of Types, Their Relative Importance and Their Length of Service

WHAT proportion of boilers in the industrial power plants are of fire-tube or water-tube design?

Which design holds the balance of installed capacity?

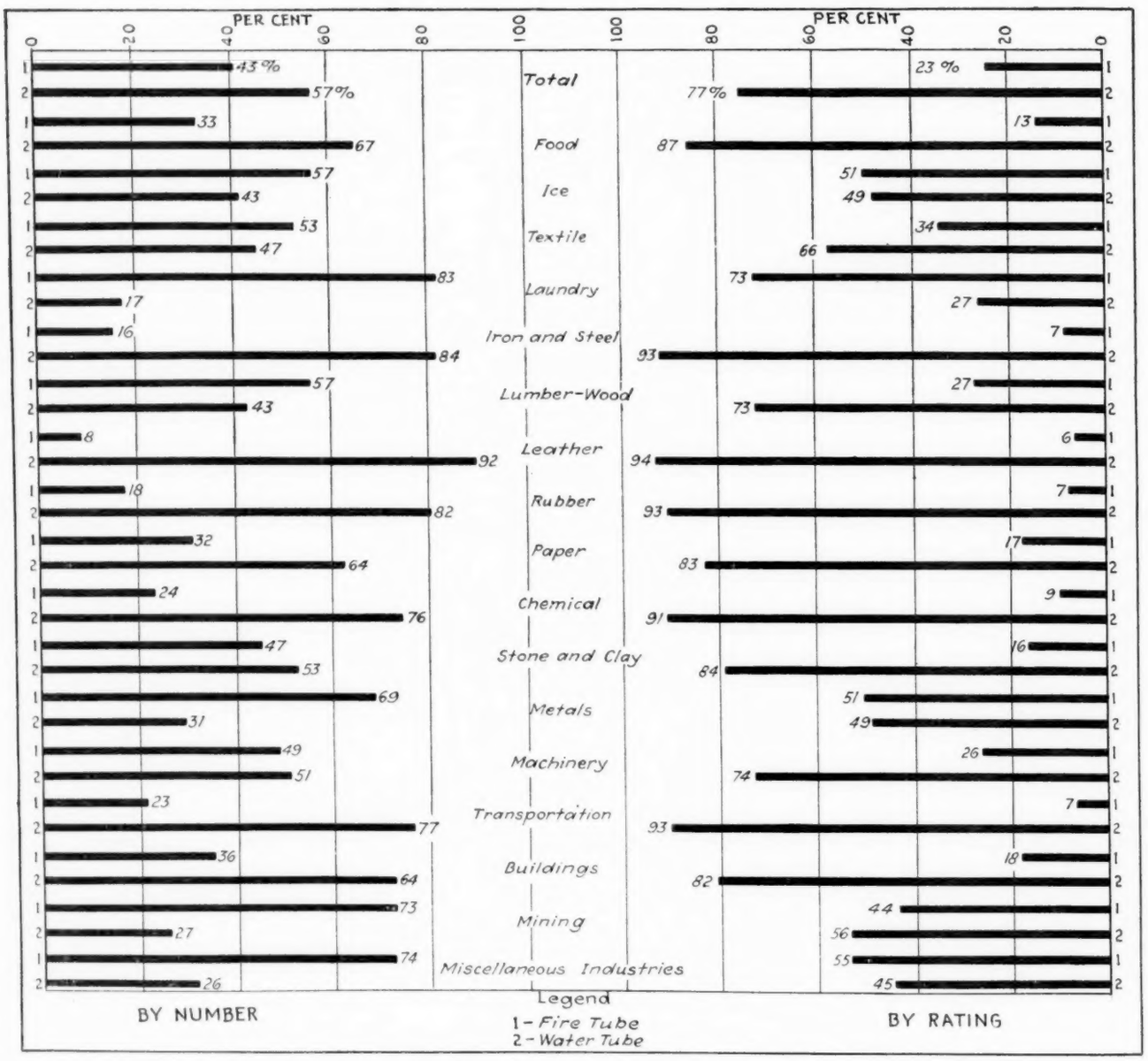
How do these comparisons change between the industries served?

How many years of service have been rendered by the boilers now installed?

The answers to these questions are given here in two tables, in terms of per cent, from data based on approximately seven hundred answers to a questionnaire sent out by *Power* to its subscribers in the industrial groups.

In the first table is shown the percentage comparison between fire-tube boilers and water-tube boilers as to numbers in service and as to rated capacity. The left-hand column contains the numerical comparison and the right-hand column the division by rating. Thus, it is seen, for all the industries combined under the heading of "Total," that fire-tube boilers represent 43 per cent of all installations, with the water-tube boilers holding 57 per cent. In the right-hand column the fire-tube boilers represent 23 per cent of the total capacity, while the water-tube boilers aggregate 77 per cent. By following down the two columns, the condition can be checked for each of the industries.

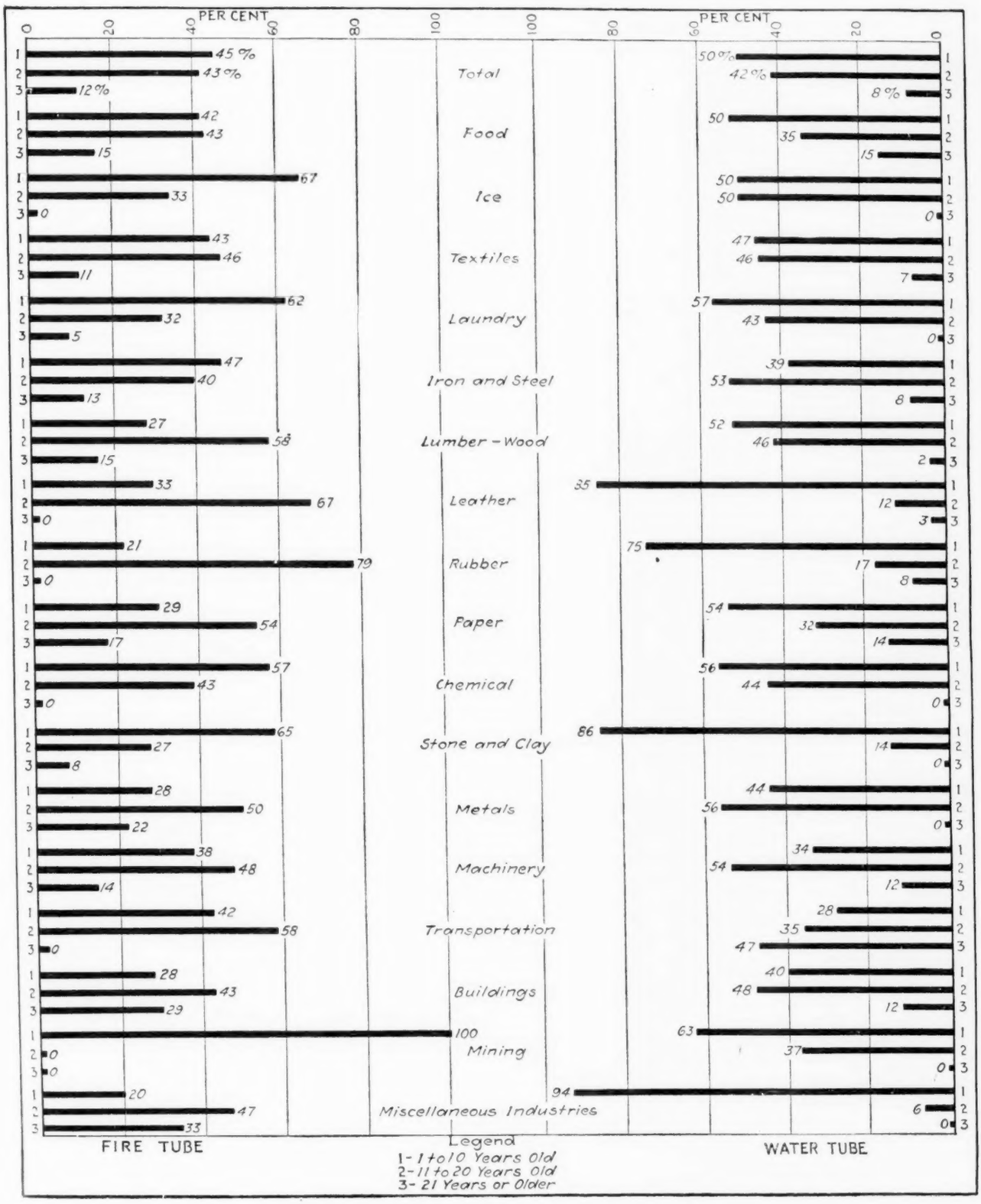
The second table shows the length of service in three


PER CENT
0 20 40 60 80 100
100 80 60 40 20 0
PER CENT
Total 43% 57% 23% 77%
Food 33 67 13 87
Ice 57 43 51 49
Textile 53 47 34 66
Laundry 83 17 73 27
Iron and Steel 16 84 7 93
Lumber-Wood 57 43 27 73
Leather 8 92 6 94
Rubber 18 82 7 93
Paper 32 64 17 83
Chemical 24 76 9 91
Stone and Clay 47 53 16 84
Metals 69 31 51 49
Machinery 49 51 26 74
Transportation 23 77 7 93
Buildings 36 64 18 82
Mining 73 27 44 56
Miscellaneous Industries 74 26 55 45
Legend
1 - Fire Tube
2 - Water Tube
BY NUMBER
BY RATING

A percentage comparison of fire-tube and water-tube boilers by numbers and by installed rating

age classifications, for fire-tube boilers to the left and for water-tube boilers to the right. It can be seen for all industries combined, at the top and to the left, that 45 per cent of the fire-tube boilers are 1 to 10 years old, 43 per cent have seen from 11 to 20 years of service and 12 per cent have been in use for 21 years or more. Similarly, water-tube boilers, when distributed in the three age classifications, show 50 per cent in the first or newest, 42 per cent in the second and 8 per cent in the 21-year or older group.

The proportion that falls in each age classification for a given industry is shown, and by checking the figures some interesting conclusions can be drawn on the status of the industries and the relation between types.



The percentage of boilers in three age classifications

New Thoughts on Supercharging Oil Engines

By ROBERTSON MATTHEWS

THE core of all discussion on the supercharging of large engines seems to be either the control of temperatures or else limitation of maximum pressure. Improvement in combustion and its possible results receive secondary attention. Hence, while over 30 per cent supercharge is declared advisable by some engineers, only 10 per cent is favored by the more conservative.

Meanwhile it is noteworthy, concerning temperature effects, that while the supercharging of 24-in. cylinders is by some regarded with caution, the successful operation of 33-in. unsupercharged cylinders has led some to vision the 40-in. cylinder and over. Now since heat effects vary not only with gas temperature but with the diameter of the cylinder — that is, with the amount of the hot gases — the real cause for the supercharging of Diesels is not at all clear.

If 40-in. cylinders lie in the offing while some engineers hesitate to supercharge 24-in. cylinders, it looks as if there were two views of the supercharging question—that of supercharging solely and that of the possibility of the advent of supercharging acting as a stimulus to further Diesel-engine development.

OPINION is divided on the merits of supercharging four-stroke-cycle Diesel engines. The items involved are many, hence results differ from forecasts, and facts seemingly fail to check with theory. Reports of supercharging cover as widely different conditions as supercharging a Diesel in order to get 30 per cent increased capacity, and that of supercharging gasoline engines to give mean effective pressures up to 185 pounds in order to eliminate the need of changed gears. The idea of supercharging during combustion is here advanced.

The pure Diesel with its injection air is already a supercharged engine. As compared with the airless injection engine, it supercharges from 7 to 10 per cent, through the admission of the injection air. Yet but little is said of this supercharging that favors the pure Diesel.

If combustion were already what it should be in Diesel and other oil engines, supercharging would not raise the temperature to such uncertain values as are feared by some. Thus

$$\textit{Temperature rise} = \frac{\textit{B.t.u. sensible heat from combustion}}{\textit{weight of gases} \times \textit{spec. heat of gases}}$$

Clearly, 30 per cent more fuel and 30 per cent more air would simply increase both numerator and denominator in equal proportion, and there would, with the same intensity of combustion with each unit of fuel, be no change in the temperature rise.

But even Diesel engines are not without some afterburning. Hence, since most fuels have a lower ignition temperature as the concentration of oxygen increases, and since combustion is improved with improved access to oxygen, the rate of combustion can vary when supercharging is introduced. This variation can change both the time when the maximum temperature will occur and also its magnitude, neither of which conditions can be definitely forecasted. On the other hand, there can be the benefits of reduction in late burning. The compression temperature can vary, of course, only with r_v, n, and the initial air temperature, t_a as shown in the table.

Regarding pressure changes, with the same air temperature and cylinder volume, the initial pressure will go up directly as the increase in weight of air introduced. The table shows how the final compression pressure p_c goes up as the initial air pressure p_a is increased. Different values of n, the exponent of the compression curve, would occur, depending on the size of the cylinder and the piston speed. The ratio of compression based on initial and final volumes is r_v; based on final and initial pressures, it is r_p, t_c being the final compression temperature.

The maximum pressures may become greater with supercharging owing in part to combustion being influenced by the reduction in the percentage of residual gases. This reduction can be due either to greater weight of air charge or to better scavenging resulting from some blow-by from the inlet to the exhaust possible in all except, probably, Tosi engines. Besides this, if the precompressed air comes to the engine with heat added by the blower and it happens that the amount of residuals is not greatly reduced, the final compression temperature could be increased, also the maximum temperature, and hence the maximum pressure.

These facts suggest that statements regarding pressure and temperature increases from supercharging, are open to qualification and may or may not lend support to a conservative attitude toward supercharging.

Regarding fuel economy, consider first no increase in the fuel injection. Earlier combustion, within limits, due to greater concentration of oxygen from better scavenging and higher compression pressure, should improve fuel economy and hence indicated capacity. Greater maximum load means greater net load, and therefore higher mechanical efficiency, but when greater capacity is desired by increasing the fuel injection, then limitations of the Diesel cycle and the demands for reliability raise, at least at the present time, serious problems if large gains from supercharging are expected. Later cutoff would reduce economy and earlier cutoff, but with more fuel injected per degree of crank travel, would increase the maximum pressure. With cutoff unchanged, but with more fuel in proportion to

the increase in air, higher pressures would always occur.

Unfortunately, the pure Diesel is essentially a lean-mixture engine, because for high economy it demands early cutoff. Compared with aëro-gasoline engines, many Diesels are low-capacity machines; and after allowing for a difference in the number of expansions of about 3 to 1 the Diesel is not so remarkable on economy. It has shone, not by comparison with a limit of excellence, but by comparison with a less competent creation, the steam engine. In fact, the pure Diesel shows only about two-thirds the ideal thermodynamic efficiency, and if to this is added the present-day weakness of all oil engines for great percentages of excess air, there is reason for questioning all conditions affecting the present-day efficiency and capacity of Diesel engines, and for pondering whether there may not be an over-emphasis on maximum pressures and large cylinders.

To reduce the compression pressure in order to gain capacity from supercharging while maintaining conservative maximum pressures, seems odd procedure. For high compression means high thermal efficiency and high thermal efficiency means high capacity from fuel. High capacity from fuel, especially when containing carbon (and Diesel fuel is over 80 per cent carbon) means high capacity from air. High capacity from air means high capacity per cubic foot of piston displacement, which is the present-day purpose of supercharging.

Hence reducing the compression ratio in order to supercharge may be likened to coming down one step in order to be able to leap up two. The excuse, of course, is reliability.

Effect on Reliability

Concerning one item in reliability, exhaust valves have been reported to suffer from supercharging. In one case attention was required in 40 days as against 60 days with no supercharging. It would appear that the more complete and therefore earlier combustion possible with supercharging would benefit exhaust valves. However, heat transfer to the valves varies with the flow of the exhaust gases; 30 per cent increase in the mass of gases would mean 30 per cent increase in flow if the temperature did remain the same with supercharging as without. The gain, however, in capacity from substituting a simple feature such as a blower in place of a portion of an engine, should allow some increase in expenditure on the material of parts.

The short running time of an aëro engine between overhauls, astounds some Diesel engineers, but consider some numerical values. Taking the 60-day interval just menticned, for the Diesel engine the number of exhaust valve openings = (60 days $\times$ 24 hours $\times$ 60 minutes $\times$ 125 r.p.m.) $\div$ 2 = 5,390,000. For an aëro engine, (225 hours $\times$ 60 minutes $\times$ 1,600 r.p.m.) $\div$ 2 = 10,800,000 exhaust-valve openings. While the Diesel has a larger valve to cool, the aëro engine has hotter gases containing, by comparison, almost no excess air, and cooled, at the time of exhaust opening, by only about one-third the expansions of the Diesel engine, to say nothing of there being later burning. The aëro engine here represented was, by the way, air cooled.

According to thermodynamics, supercharging should logically be accompanied by the addition of an exhaust-gas turbine to do the precompressing, since without this gas turbine the ratio of compression has been increased without compensating for it with increased expansion.

When no air is considered for additional scavenging due to blow-by, the work to precompress the air may be taken roughly as the friction load of blower and the machine driving it. For the pressure built up in the blower becomes positive pressure on the engine piston during charging, but would show only on a weak-spring card. Refined comparisons of the indicated horsepower with and without supercharging should consider this lower loop effect, which could amount to a flat 5 lb. difference in the mean effective pressure. This partial balancing of the blower work would seem to contradict the foregoing statement that the engine should have increased expansion to compensate for increased compression.

Work Gained by Precompression

This would be so if the engine gave back only the effort put into it as work. However, the function of an engine is to put out effort in return for energy put into it at least. Hence, this precompression should, strictly, go to aid the engine to utilize heat through the work of expansion, and to an amount greater than the work of compression, depending on the cycle efficiency.

Summing up the influence of supercharging on fuel economy, if supercharging is considered as a substitute for using a later cutoff to gain capacity, then increased economy can result from the more favorable cutoff and also due to earlier and better combustion. If the compression ratio is reduced in order to keep down maximum pressures, then the reduced expansion opposes the gain from better combustion due to more air.

Concerning supercharging as a means of controlling temperature while gaining capacity, it is significant that supercharging trials with a gas engine gave 20 per cent increase in m.e.p., yet showed no temperature rise during the cycle. Evidently, there was early and complete combustion both with and without supercharging. This raises the question as to how far heat troubles when using liquid fuel depend upon the mass of the hot gases radiating heat and upon the length of time during which burning occurs.

High Temperatures Not Necessary

High as is the rate of radiation from high-temperature gases, the heat dispersion during the molecular vibrations of combustion is declared to be much more intense. Hence, the heat loss during late burning becomes a matter not alone of the temperatures supposed to exist on the expansion curve, but of something more intense in its heat dispersion. Accordingly, reports of reduced jacket loss with supercharging appear reasonable if supercharging improves combustion even without leaning the fuel-air ratio.

Whereas, the conception of supercharging controlling temperature effects by dilution of the gases, puts supercharging somewhat in the class of jacket-water control, control of heating by improving combustion and reducing late burning raises supercharging to the plane of real engine development. The question then arises as to how best to use extra air in order to get the most satisfactory combustion.

Since much benefit to combustion is attributed to the mixing action of injection air, more turbulence would appear to be helpful up to the point where detonation impends. A compromise could therefore be effected

between gain from excess air and from turbulence. Such supercharging during combustion calls for considerable work to compress this air above combustion pressures. On the other hand, increasing the air charge during intake increases the work of total compression. Ignoring, therefore, the difference in the mechanical efficiency of the engine as a compressor and of a separate compressor, the difference in the work of compressing the supercharging air becomes that corresponding to the additional pressure head necessary above combustion pressure. In return for the extra work due to this head, there is the advantage of being able to control the supercharging air to give directed turbulence.

Such a system presents greater mechanical complexity than precompressing the entire air charge, but it would seem that this directed turbulence could also be used to throw a curtain of cooler air across, say, an exhaust valve and thereby save some heating by conduction. For contrary to a common view, conduction is said to take on rates of heat transfer in greater proportion than the first power of the temperature when temperatures corresponding to combustion occur.

Such a system should be particularly applicable to high-speed Diesels. But as with higher pressures, possibly nothing definite will appear till a new balance is struck between competition and investment efficiency.

This discussion will appear more applicable to the pure Diesel than to the airless injection engine. Since the latter engines are, however, increasing not only in number but in size, their supercharging deserves attention. Having greater clearance volume than the pure Diesel, together with higher pressures, it would appear

EFFECT OF HIGHER INITIAL PRESSURES

	n, value of exponent in $PV^n = K$					
	1.4	1.38	1.36	1.34	1.32	1.3
r_p	15	15	15	15	15	15
t_c*	1,340	1,250	1,157	1,070	992	913
r_p	44.2	42	39.8	37.7	35.7	33.8
P_a	P_c, Final Compression Pressure					
14	618	588	557	528	500	473
16	707	672	637	603	572	546
18	796	756	717	678	643	608
20	883	840	796	754	714	676
When * $r_p = 12.5$ $t_c = 1,215$		1,133	1,055	980	908	843

that the airless injection engine should have a lower volumetric efficiency. Add to this the absence of any injection air, and the solid-injection engine probably falls 10 per cent short on air supply compared with the pure Diesel. Besides this, there is the poorer atomization and distribution. Hence the airless injection engine would seem entitled to at least some supercharging.

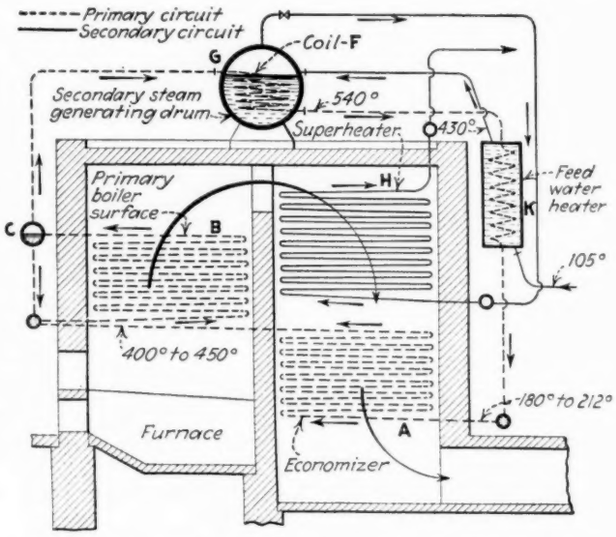
But with only rare exception, the maximum pressures of the airless injection engine exceed considerably those of the pure Diesel without supercharging. Furthermore, once combustion has started, there is with solid injection no restraining influence of cold injection air. Given, therefore, greater concentration of oxygen and probably some increase in turbulence, the uncertainty of maximum pressures should suggest caution in advancing initial pressures, whether or no the volumetric compression ratio is changed.

Sensitiveness to attempts at supercharging during combustion would be greater for the airless injection engine than the pure Diesel because of the greater inherent absence of turbulence in most airless injection engines.

Steam Generated Indirectly in German Experimental Boiler

A test boiler designed for operation at 900 lb. per sq.in. pressure and for the indirect production of steam in a drum equipped with heating coils, has recently gone into experimental operation in Germany. According to the March 17 issue of the *V.D.I. Nachrichten* (newspaper of the Society of German Engineers), this boiler was installed by the Schmidtsche Heissdampf-Gesellschaft in their laboratory at Wernigerode to make possible the study of heat transfer and similar problems as they apply to high-pressure steam.

The accompanying diagram shows the method of operation. There are two fluid circuits, the primary circuit starting with water entering the economizer *A* at a temperature around 200 deg. F. From the economizer it passes to the evaporating tubes *B*, subjected directly



Fluid in the primary circuit A, B, C, F, K, A, never leaves the boiler plant

to the heat of the fire, then through a small drum *C* where most of the steam is disengaged from water. The steam then passes to the heating coil *F* in the large drum *G*, evaporating practically its own weight of the surrounding water and being itself condensed in the process. The condensate passes through the feed-water heater *K*, where it gives up some of its heat to the incoming feed water of the secondary circuit. It then passes back to the economizer *A*.

Since this primary circuit is completely closed, no makeup is necessary and no trouble should be experienced from scale in the economizer or primary boiler surface. This is apparently one of the reasons for the design shown, the other being the elimination of a large drum weakened by tube connections.

The feed water of the secondary circuit passes through the feed-water heater *K* and then goes to the evaporating drum *G*. The steam produced by the heat of the coil *G* passes out through the superheater *H* to the point of use. While much priming may be expected in the primary circuit on account of the small size of drum *C*, this would not appear to be objectionable.

The removal of scale forming on the outside of coil *F* or the prevention of its formation would be determined entirely on the basis of boiler capacity.

Analyzing the Power-Purchase Problem

Factors Affecting a Decision as to Purchasing or Generating Power for Industrial Plants, Particularly Those Requiring Steam for Heating and Process Purposes

By ALBERT C. WOOD
Consulting Engineer, Philadelphia, Pennsylvania

WHETHER it will be more advantageous for an industrial establishment or a public or private institution to purchase power or to generate it in an existing plant or in some proposed new plant, is a question that has arisen many times in the past, and one that is being considered with increasing frequency, as a result of the rapid growth and extension of the large central-station power business in this country.

Many factors must be considered in arriving at a proper solution, some of these factors being economic, some strictly financial and some engineering. None of these considerations can be overlooked by the management, although the engineer may be asked to report upon the engineering phases only, including construction and operating costs and advantages under the purchased power and private plant plans.

Not infrequently, the credit condition and borrowing capacity of the owners and the need and value to them of money for extending the producing or serving ends of the enterprise, will dictate a solution of the problem which may be quite contrary to that which would be arrived at, under other circumstances, by a careful analysis of total operating costs under the two plans.

Too often, managements will postpone action until power difficulties become acute, and frequently a decision will be required before a proper investigation and analysis can be made. The difficulties and cost of investigations are often greatly increased by a paucity of operating records, making it necessary to conduct tests of a more or less elaborate and expensive nature, in order to obtain the required information with which to work.

Principal Factors Summarized

The principal factors to be considered in arriving at a decision concerning the purchase of power may be briefly summarized as follows:

1. Characteristics of Industry:
   - (a) Product manufactured, or business served.
   - (b) Stability of industry and future possibilties.
   - (c) Working hours.
   - (d) Requirements for heating and process steam (if any) and adaptability of low-pressure steam to process.
   - (e) Waste fuel or waste heat (if any) from which steam and power may be generated.
   - (f) Vital processes and conditions affecting importance of continuity of service.
   - (g) Relative efficiency of electrical and mechanical drives for particular service.
   - (h) Existing d.c. or a.c. installations and advantages of each for the industry in question.
   - (i) Earnings of capital in industry as compared with earnings of capital invested in power plant.
2. Load Characteristics:
   - (a) Electrical or mechanical loads, for different hours of day and for typical weekday, Saturday afternoon, Sunday and holiday, during different periods of the year.
   - (b) Process steam (or hot water) and heating loads for like times and periods.
   - (c) Abnormal lighting and power loads and their effect on peak demands.
   - (d) Coincidence of power loads and process and heating requirements for different periods of year.
   - (e) Power factor of load and possibilities of improvement.
3. Existing Power Plant and Service System:
   - (a) Character, adequacy and efficiency of existing plant for supplying present requirements for power, and (or) heating and process steam.
   - (b) Probable economic life.
   - (c) Adaptability for improvement and extension.
   - (d) Available space for extension.
   - (e) Suitability of location with reference to present industry and future extensions, sidings and coal storage, water-supply sources, etc.
   - (f) Dust and dirt nuisance of present location.
   - (g) Nature and extent of existing service system and adaptability for future conditions.
   - (h) Value of site for manufacturing purposes.
4. Available Site for New Plant:
   - (a) Location with reference to center of load.
   - (b) Adequacy of plot for present and future needs.
   - (c) Value of site for manufacturing purposes, or cost of new site.
   - (d) Railroad facilities and siding possibilities.
   - (e) Suitable water for condensers and boiler feed.
   - (f) Ash disposal.
   - (g) Location of plant with reference to dust and dirt nuisance.
5. Available Fuels and Fuel Costs:
   - (a) Fuel best adapted to plant conditions and adequacy of supply.
   - (b) Present cost of such fuel and probable future trend of prices.
   - (c) Waste fuels (if any).
   - (d) Waste heat (if any): Quantity, analysis and temperature of waste gases; variations in temperature of gases and rate of production, dust conditions, etc.
6. Available Water Supplies:
   - (a) Adequacy and suitability for condensing purposes (if required) and for boiler feed.
   - (b) High and low water conditions.
   - (c) Débris, mud and scale-forming matters, etc.
   - (d) Requirements for pumping.
   - (e) Temperature of supply—summer and winter.
   - (f) Cost of rights of way and riparian rights, if same must be purchased.
   - (g) Underground water resources (if any).
7. Labor Conditions and Rates:
   - (a) Character and dependability of available labor.
   - (b) Labor rates for skilled and unskilled employees.
8. Purchased Power Facilities:
   - (a) Character and adequacy of plant and transmission system.
   - (b) Distance and hazards of transmission.
   - (c) Record of service interruptions in immediate locality, for past year or two, voltage variations, etc.
9. Purchased Power Rates Applicable to Service:
   - (a) Demand charges.
   - (b) Minimum guarantees (if any).
   - (c) Energy rates.
   - (d) Coal clauses.
   - (e) Power factor clauses.
   - (f) Probable future trend of purchased power costs.

Estimated Power Plant Operating Conditions and Performance

New Power Plant vs. New Boiler Plant and Purchased Power for Year 1933-34

Based on Study of Actual Case, Assumed as Typical

Season	No. of Months	Period	Power and Lighting: Hours per Year	Power and Lighting: Av. Factory Load, Kw.	Power and Lighting: Max. 15 Min. Demand, Kw.	Power and Lighting: Power to Station Auxiliaries, Kw.	Steam Distribution: Steam for Processes and Heating Buildings: Lb. per Hr.	Steam for Processes and Heating Buildings: Press. Lb.	Steam Distribution: Turbines in Operation E.T. Extract. Type N.C. Non Cond Type	Steam to Turbines, Lb. per Hr.	Equiv. Steam Charged to Motor-Operated Auxiliaries	Steam to Boiler Feed Pumps	Steam to Maintain Press. in Distrib. System, Lb. per Hr.	Steam to Condensation Leakage, Soot Blowing and Misc. Lb. per Hr.	Av. Steam Supplied By Boilers, Lb. per Hr.	Max. Steam Demand, Lb. per Hr.	Boiler Performance: No. of 6,000 Sq.Ft. Boilers in Operation	No. of 6,000 Sq.Ft. Boilers Banked	Av. Per Cent Boiler Rating of Boilers on Load	Per Cent Boiler Rating on Peak	Efficiency (Excluding Banking) Per Cent	Coal Consumption: Av. Coal per Hr. (Excluding Banking)	Coal for Banking, Etc., Lb. per Hr.	Total Coal per Hr.	Gross Tons Coal Per Year
All Steam and Electrical Power Generated in Private Plant																									
Season	2½ Mos.	Day	630	1,651	3,200	113.0	14,000	20	2-1500 E.T.	41,000	2,500	1,350	.....	6,750	51,600	81,000	3	0	93	145	72.5	5,570	.....	5,570	1,568
		Night	680	256	425	78.5	.....	.....	1-1500 E.T.	5,640	1,990	1,000	2,000	4,450	15,080	16,000	1	2	81	86	71.2	1,654	250	1,904	578
		Sat. P.M.																							
		Sun. & Hol.	515	55	120	7.5	.....	.....	1-150 N.C.	3,080	425	350	.....	3,750	7,605	14,000	1	0	41	75	62.0	960	70	1,030	237
Warmer	4½ Mos.	Day	1,151	1,737	3,200	115.0	17,700	20	2-1500 E.T.	44,200	2,520	1,420	.....	6,750	54,900	93,000	3	0	98	167	73.2	5,800	.....	5,800	2,980
		Night	1,245	890	1,425	91.0	5,200	10	1-1500 E.T.	18,500	1,550	1,100	1,200	5,050	27,400	45,000	1	2	147	242	72.3	2,970	250	3,220	1,790
		Sat. P.M.																							
		Sun. & Hol.	889	55	120	7.5	4,230	10	1-150 N.C.	3,080	425	350	1,500	3,750	9,105	14,000	1	0	49	75	65.0	1,097	70	1,167	463
Colder Season	5 Mos.	Day	1,266	2,175	3,200	152.5	59,800	20	1-1500 E.T. 1-1500 N.C.	72,000	2,450	1,400	.....	7,500	83,350	118,000	4	0	112	159	74.0	8,800	.....	8,800	4,962
		Night	1,373	1,160	1,425	116.5	44,800	10	1-1500 E.T. 1-1500 N.C.	47,600	1,920	1,200	.....	6,000	56,720	79,000	2	2	154	212	72.3	6,140	250	6,390	3,920
		Sat. P.M.																							
		Sun. & Hol.	1,011	68	140	51.5	44,800	10	1-150 N.C.	2,920	2,225	1,130	39,650	4,500	50,425	66,000	2	0	135	177	73.8	5,340	70	5,410	2,440
Totals and Weighted Av.			.....	8,812,470 Kw.-Hr.	.....	785,720 Kw.-Hr.	.....	.....	.....	.....	.....	.....	.....	.....	44,000	.....	.....	.....	109.5	.....	71.4	.....	.....	.....	18,938
Steam for Process and Heating Generated in Private Plant—All Power Purchased from Power Company																									
Season	2½ Mos.	Day	630	1,651	3,200	32.5	13,650	200	.....	.....	.....	800	.....	6,750	21,200	26,000	1	0	106	132	74.0	2,125	.....	2,125	598
		Night	680	256	425	4.0	.....	.....	.....	.....	.....	400	2,000	4,450	6,850	7,850	1	0	35	40	58.0	876	.....	876	266
		Sat. P.M.																							
		Sun. & Hol.	515	55	120	3.0	.....	.....	.....	.....	.....	350	2,000	3,750	6,100	7,360	1	0	31	38	57.0	848	.....	848	195
Warmer	4½ Mos.	Day	1,151	1,737	3,200	35.5	16,920	200	.....	.....	.....	885	.....	6,750	24,555	41,000	1	0	125	208	74.0	2,460	.....	2,460	1,264
		Night	1,245	890	1,425	28.5	5,050	200	.....	.....	.....	530	1,200	5,050	11,870	30,000	1	0	61	153	67.5	1,303	.....	1,303	725
		Sat. P.M.																							
		Sun. & Hol.	889	55	120	15.0	4,100	200	.....	.....	.....	350	1,200	3,750	9,400	29,600	1	0	48	151	63.0	1,107	.....	1,107	439
Colder Season	5 Mos.	Day	1,266	2,175	3,200	52.0	58,200	200	.....	.....	.....	1,200	.....	7,500	66,900	82,000	3	0	114	139	74.0	6,710	.....	6,710	3,795
		Night	1,373	1,160	1,425	44.0	43,600	200	.....	.....	.....	1,100	.....	6,000	50,700	62,000	2	1	129	158	74.0	5,090	125	5,215	3,200
		Sat. P.M.																							
		Sun. & Hol.	1,011	68	140	42.0	43,600	200	.....	.....	.....	1,100	.....	4,500	49,200	60,500	2	0	125	154	74.0	4,940	70	5,010	2,260
Totals and Weighted Av.			.....	8,812,470 Kw.-Hr.	.....	283,000 Kw.-Hr.	.....	.....	.....	.....	.....	.....	.....	.....	31,220	.....	.....	.....	93	.....	69.7	.....	.....	.....	12,742

While the points here enumerated are important, they constitute only a brief and more or less incomplete summary of the factors that must ordinarily form the groundwork for the engineer's analysis and recommendations.

Limitation of space will preclude a full discussion of the subjects mentioned, their importance, and the methods that may be pursued in securing and analyzing the various data. The latter must largely be left to the ingenuity, judgment and experience of the engineer.

Load Data, Present and Future

A thorough knowledge of the present load conditions and of the power situation in general is most essential; quite as important, in many cases, is a reasonably correct picture of the future. Sometimes the expansion program of an industry is carefully mapped out for several years ahead, and in such cases it is not so difficult to determine about the added steam and electrical loads and to lay out load curves for the future by superimposing new loads on the curves applying to present-day conditions.

Use Factors Applicable to Exhaust Steam

Where provisions are not made in an existing plant for securing a balance between low-pressure steam production and utilization, the proportion of exhaust actually used to that which is produced and available, should be carefully studied, in order that an accurate estimate can be made of the live steam equivalent of the exhaust used, or the amount of steam that would have to be bled from turbines. Often, the importance of this point is overlooked, and assumptions are made that are not in accordance with the facts.

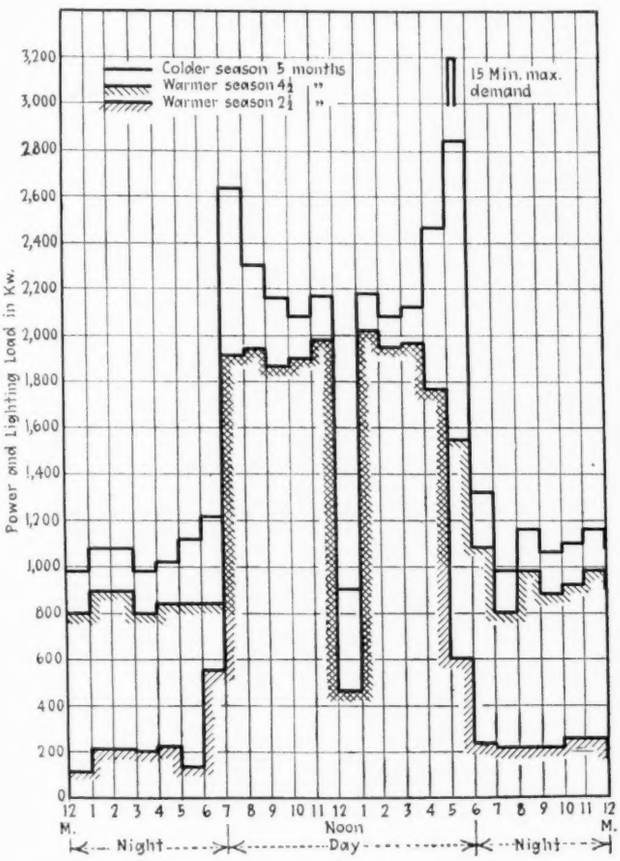


Fig. 1—Typical daily load curve for different seasons

Mechanical Drives and Motor Installations

In plants where the power applications are effected through mechanical drives, or direct-current motor installations, which it would be desirable or necessary to replace with alternating-current motor equipment in case power were purchased, the advisability and cost of making the change may be the subject of an investigation quite as important as that of the source and cost of the power supply. The inquiry may involve the consideration of an immediate and complete change-over, or one to be accomplished in stages, over a period of years, with possible employment, in the interim, of motor-generator sets or large group drives, for the operation of sections of the industry.

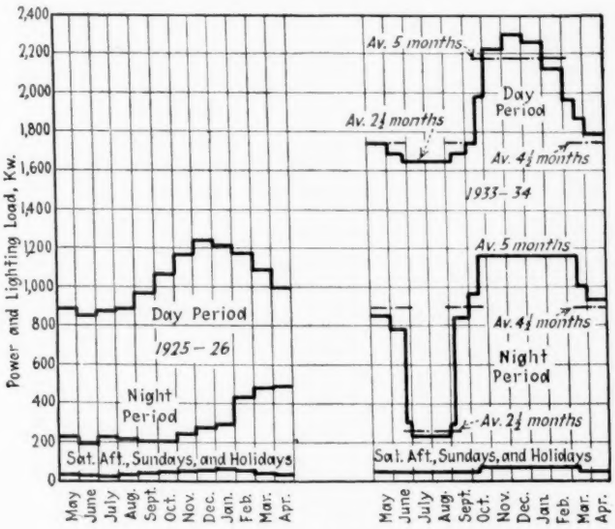


Fig. 2—Monthly electrical load curves for 1925-26 and estimated for 1933-34

In some cases the replacement of engines and mechanical drives may be impossible to justify, on account of the efficiency and reliability of the power application and the low fixed charges and cost of maintenance. The same thing may also be true with certain direct-current power installations. Unless some tangible and worthwhile advantage can be shown for making a change, managements will seldom be willing to invest the money for them.

Service System

The character and extent of the service system connecting any existing plant with the industry which it serves, and the nature of the changes and additions required for future conditions, whether the existing plant is extended or a new plant is built, may prove important considerations. Frequently, the cost of the service system under two or more plans, may be quite different, and one or the other may involve relatively large expenditures. This is especially true if the services are to be carried underground and tunnels provided between the power house and the various buildings.

In many industries dependability of power service is of first importance. It is especially important to industries employing a large number of machine workers or to

those carrying on processes of a more or less vital character. The managements of such industries will often be loath to tie up completely to purchased power, especially if the transmission is a long one. In their own plants, they will insist on ample reserve to provide for proper maintenance and insure against breakdowns. In such cases, if not in all cases, the engineer should make a careful inquiry into the number, duration, time of occurrence and cause of the service interruptions, which have happened in the immediate locality in the past two or three years, and ascertain what, if any, steps have been taken to improve the service. Also, it would be well to ascertain the loss resulting to the industry, per hour, from day and night interruptions.

Reserve Capacity Essential

In a private plant, operating an industry for which continuity of service is of prime importance, ample reserve capacity must not only be provided in the plant, but must be ready for instant service, at least during the important working hours of the day. Such plants must sacrifice some measure of efficiency to dependability of service.

At least one reserve unit should be provided to allow for proper maintenance. The keeping under steam of only one boiler in a plant, and the operation of this boiler at high ratings, is a risky practice and one to be condemned when continuity of service is important or the feed-water conditions are not the best.

Reserve capacity in electrical generating equipment is, in many cases, not as important as reserve capacity

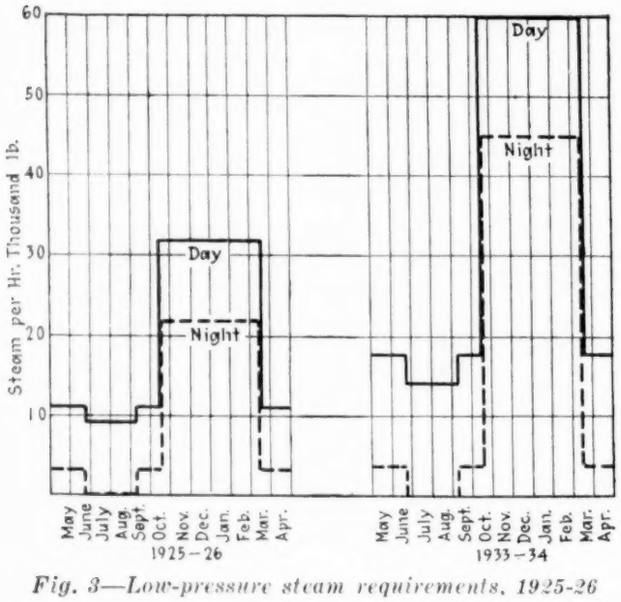


Fig. 3—Low-pressure steam requirements, 1925-26 and 1933-34

in boilers. Certain unessential load may sometimes be dropped, making it unnecessary to keep in service more than two generating units. A third unit should, however, be provided in the plant, unless the peak load, in an emergency, can be carried on one unit. Two generating units may suffice for industries in which certain operations carried on during the day can temporarily be transferred to the night period.

Capital Expenditures and Fixed Charges

In making up his estimate of the cost of a new power plant or of necessary additions to an existing plant, the engineer should regard the "economic balance" between expenditures and capitalized economies as second only in importance to "dependability." A plant to serve an industry working only 10 hours a day, or, say, 55 hours a week, and located where the fuel supply is reasonably cheap, may and should be of a very dif-

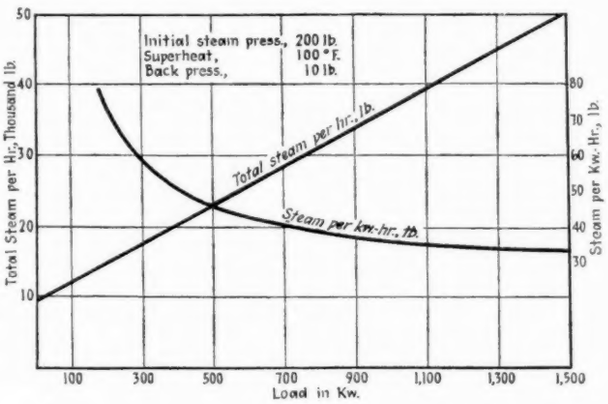


Fig. 4—Steam consumption curves for 1,500-kw. non-condensing turbo-generator

ferent character from the plant that must operate 24 hours per day for 6 or 7 days in the week. If the private plant operating, ordinarily, 9 or 10 hours a day, is to compete with purchased power, it cannot be burdened with unessentials. On the other hand, 24-hour service and a high load factor will warrant a more expensive and efficient plant.

Often, a new plant will be expected to carry not only its own fixed charges, but also those applying to an unamortized investment in an existing plant. In such case any plan, such as purchased power, set up in comparison with the new plan should carry fixed charges on the outstanding old investment and on the new fixed capital as well.

The rate at which depreciation should be figured, and interest and other allowances entering into the item of fixed charges, will depend on the character and stability of the business, its credit standing, etc. Ordinarily, fixed charges for industrial enterprises may be figured at 12 to 15 per cent.

Basis of Comparison of Operating Costs

Where process and heating steam, in addition, to power, are supplied from the power plant, comparisons of operating costs, when using purchased power, should always be made on the basis of total annual operating cost, including fixed charges. Cost comparisons on the kilowatt-hour basis, in such cases, are difficult to arrive at and should be used cautiously, as the allowance that should be made for low-pressure steam is one that is open to much discussion and various interpretations.

Hypothetical Case Analyzed

A hypothetical case may serve to bring out clearly some of the points discussed and the methods by which comparative operating costs under the purchased power and private plant are worked out. The conditions assumed by way of illustration are, in large part, those which actually obtained with a certain large industry in the neighborhood of Philadelphia. The principal factors in the case may be summarized as follows:

(a) Electrical energy for power and lighting, and steam for heating and certain process purposes is to be supplied to a group of manufacturing buildings, extending along a main-line railway for a distance of about half a mile.

(b) Live steam for heating and process work now supplied from an old and relatively inefficient steam plant, located along the railroad about 700 ft. from the center

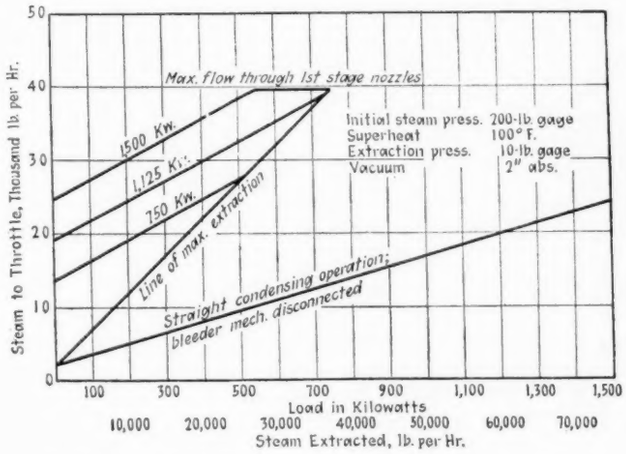

Steam to Throttle, Thousand lb. per Hr.
Max. flow through 1st stage nozzles
1500 Kw.
1,125 Kw.
750 Kw.
Line of max. extraction
Straight condensing operation; bleeder mech. disconnected
Initial steam press. 200-lb. gage
Superheat 100° F.
Extraction press. 10-lb. gage
Vacuum 2" abs.
Load in Kilowatts
Steam Extracted, lb. per Hr.

Fig. 5—Steam-consumption curves for 1,500-kw. extraction type turbo-generator

of distribution, the pressure being reduced to various buildings to approximately 5 lb. for heating and 15 lb. for process work.

(c) Power for all purposes purchased.

(d) Working hours of industry, ordinarily 9 to 10 hours, but hours vary in different departments, some working at night during 5 colder months, and others for certain period at night during 9½ months.

(e) Existing power plant inadequate for future conditions and antiquated.

(f) Satisfactory site for new power plant adjoining present plant, the location being favorable as regards coal siding and storage facilities, and water for boiler-feed purposes, but unfavorable with respect to adequate, natural water supply, necessitating the use of spray ponds or cooling towers for condenser supply.

(g) Records of electrical power consumption for several years past, available, also satisfactory data in regard to low-pressure steam use for heating and process work.

(h) Expansion program for next five years definitely laid down.

(i) Problem is to determine whether it would be advisable to continue to operate with purchased power or to build a new plant for supplying all electrical and steam service, having in mind that a new boiler house will be necessary in any case, and that this should be of a character and design suited for use in connection with a complete, steam-electrical plant, in case, later on, it should be found advisable to discontinue the purchase of power.

(j) In order that estimates of relative operating costs under the two plans will more nearly represent the average condition which will obtain during the useful life of the plant, operating costs are to be figured on conditions assumed to obtain 7 or 8 years hence.

Load Curves—From the available data regarding present and future electrical demands, load curves of typical weekdays, for three different seasons of the year were developed. These curves are shown in Fig. 1. On these curves the estimated 15-min. monthly peak is indicated, this being estimated at 3,200 kw.

Curves were also developed to show the average load for the day and night periods, and for Saturday afternoons, Sundays and holidays, for the different months of the year 1925-26 and of the year 1933-34 (see Fig. 2). In addition, curves were developed showing the average hourly requirements for low pressure steam for the day and night periods of the different seasons of the year 1925-26 and the year 1933-34.

Determining Annual Coal Consumption—With these curves as a basis, the table (see page 778) was developed. This table shows the average and peak electric load conditions, for different seasons of the year 1933-34, and the estimated steam distribution, boiler performance and coal consumption for the several seasons, under both the private plant and purchased power plans.

Turbine Units—Figs. 4 and 5 show, respectively typical curves for the steam consumption of 1,500 kw. units of the non-condensing and extraction types, selected as being best suited for meeting the average and peak electric load conditions and for supplying the required process steam. One non-condensing unit and two units of the extraction type were determined upon, as offering the best and least expensive working arrangement. This arrangement provides 50 per cent spare capacity in turbine equipment, this reserve being regarded as necessary on account of the importance of continuity of service. The plant design provides space for an ultimate installation of four turbine units and 1,500 c.f.m., 2-stage compressors.

Boiler Units—The data of the table were also used in determining the capacity and type of boiler units. Estimates of cost were based on five 6,000-sq.ft. boilers for the private plant, and four of the same size for the new boiler house under the purchased power plan, these boilers being built for 225 lb. working pressure, furnished with superheater for 100 deg. superheat (except in case of boilers for the purchased power plan) and being equipped with underfeed stokers suitable for normal ratings of 125 to 175 per cent and peaks of

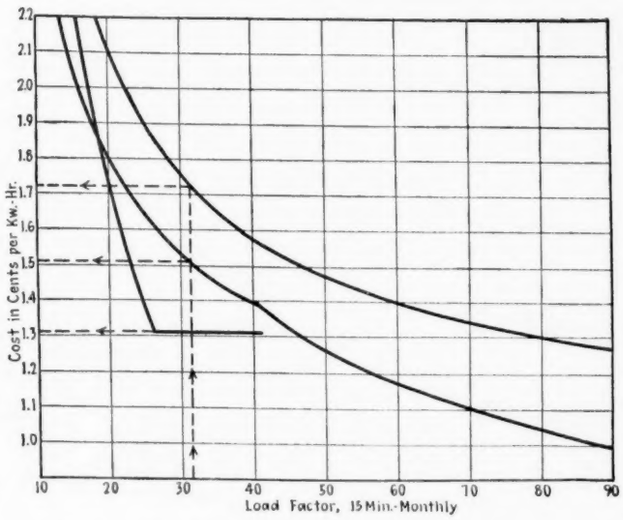


Fig. 6—Typical rate curve for purchased power

250 per cent. Boiler-house estimates in the two cases contemplate ultimate installations of six and five units, respectively. No economizer or preheater equipment is provided, since the low load factor will not justify it. Furthermore, as dependability of service is essential in this case, it is considered that the money could best be invested in spare boiler capacity.

Purchased Power Rates—Typical rate curves, applying to a 15-minute monthly demand of 3,200 kw., are shown in Fig. 6. The middle curve is assumed to apply to the case in question. The other two curves are

submitted to show differences in rates of other public-service companies operating in near-by territory, and how one is designed to secure low load factor business, while the other is a relatively poor rate under all load conditions.

Estimated Costs for Complete New Power Plant and Boiler House—In the case of the two plans under consideration, preliminary designs and estimates of cost were made. The estimated costs are summarized as follows:

COMPLETE NEW POWER PLANT

Land, clearing of site and demolition	$24,240
Railroad siding extension, track hopper and crusher house	10,500
Power house building of concrete, steel and brick, having 18 ft. basement height and consisting of boiler room 125x46 ft., heater and pumproom and storeroom 120x17 ft., turbine room 123x42 ft. and switchboard room 78x21 ft., building and machinery foundations, coal bunker, ashes hopper, suction wells, plumbing, power and light wiring, chimneys, flues, ashes tank and power-house crane	302,600
Boilers, stokers, coal- and ash-handling equipment, forced-draft fans, boiler setting and air ducts	189,840
Turbo-generators, surface condensers, air compressor, feed-water heaters and pumps, tanks, piping and pipe covering, gages, flow meters and miscellaneous	236,950
Electrical equipment, switchboard and switching equipment, bus and switch structrues, auxiliary transformers, etc.	58,000
Spray pond and spray piping system	29,460
Outside service system, including steam, water and air piping and electrical distribution system to load centers at various points	83,500
Engineering, supervision, legal, clerical and miscellaneous expense, and interest on capital invested during construction	111,550
Total	$1,046,640

NEW BOILER PLANT IN THE CASE OF PURCHASED POWER

Land, clearing of site, demolition, etc.	$17,435
Railroad siding extension, track hopper and crusher house	10,500
Boiler-house building of concrete, brick and steel, having 18 ft. basement height and consisting of boiler room 105x46 ft. and heater and pumproom 101x21 ft., building and machine foundations, coal bunker, ashes hopper, plumbing, power and light wiring, chimney, flues and ashes tank	157,900
Boilers, stokers, coal- and ash-handling equipment, forced-draft fans, boiler setting and air ducts	139,750
Heaters and pumps, tanks, air compressor, piping and pipe covering, gages, flow meter, etc.	69,750
Outside service system, including steam, water and air piping connections, new transmission lines and transformers	19,120
Engineering, supervision, legal, clerical and miscellaneous expense and interest on capital during construction	45,185
Total	$459,640

Estimated Costs of Operation—The total annual costs of operation under private plant and purchased power plans are summarized as follows:

OPERATING COSTS UNDER PRIVATE PLANT PLAN

Fuel, 18,938 gross tons semi-bituminous coal, containing 13,800 B.t.u. as received at $5.79 per ton delivered	$109,650
Storing and reclaiming coal	2,265
Ashes disposal (cartage only) at $0.50 per net ton	1,130
Operating labor	19,755
Raw and softened water supplies	4,450
Oil, waste and other supplies	1,700
Current maintenance and repairs	18,000
Fixed charges (including 7 per cent interest) on new investment	125,600
Interest on working capital at 6 per cent	2,100
Technical supervision, administration and accounting	5,000
Total Annual Operating Cost	$289,650

OPERATING COSTS UNDER PURCHASED POWER PLAN

Fuel, 12,742 gross tons semi-bituminous coal, containing 13,800 B.t.u. as received at $5.79 per ton delivered	$73,785
Storing and reclaiming coal	1,530
Ashes disposal (cartage only) at $0.50 per net ton	765
Operating labor	12,080
Raw and softened water supplies	1,690
Power for forced-draft fans, stokers, coal conveyor, pumps, lighting, etc., 283,000 kw.-hr. at $1.67	4,730
Oil, waste and other supplies	1,000
Current maintenance and repairs	11,300
Fixed charges (including 7 per cent interest) on new investment	55,160
Interest on working capital at 6 per cent	2,100
Technical supervision, administration, purchasing, etc.	2,500
Cost of purchased electric power, 8,812,470 kw.-hr. at $1.51	133,070
Total Annual Operating Cost	$299,710

The foregoing estimate of operating costs indicates an advantage in favor of the private plant plan of $10,060 per annum over and above interest charges at the rate of 7 per cent on the difference in invested capital ($587,000) in the two cases.

If power had to be purchased on the basis of the highest of the three rates shown in Fig. 6, the difference in favor of the private plant would be $29,230 per annum, and if purchased on the basis of the lowest rate shown, there would be an advantage of $8,185 per annum on the side of purchased power. If the existing plant and service system were adequate and suitable for supplying steam for heating and process purposes, making the new boiler house and service additions unnecessary, the advantage would be largely in favor of purchased power. Again, if the existing plant were of a relatively modern and efficient type, requiring only to be extended sufficiently to meet the increased load conditions, the advantage would probably be largely in favor of the private plant. The possibility of putting an old plant in more efficient condition should always be considered, and comparisons of performance with and without purchased power should be made on this basis.

Other Factors Likely to Influence Decision

In the assumed case the difference in annual operating costs is relatively small, and a decision to build a complete new plant and generate all power required, or to construct a new boiler house only and buy power, would probably depend more on other considerations than on relative operating costs. On the one hand, the management might be influenced to build a complete new plant by the greater security and freedom from difficulties of service which they would enjoy in the control and operation of their own plant, this depending largely on the transmitted power conditions; on the other hand, the added capital requirements for the complete new plant and other considerations might determine the case in favor of purchased power, in spite of the fact that operating costs under this plan might be considerably greater.

It should be borne in mind that the estimates given, on which fixed charges are based, provide for buildings and other facilities, to permit of certain future growth, and the costs are consequently higher than they would be for a plant designed to meet only the immediate needs. Further, it should be kept in mind that the cost estimates provide for service other than power, and also for important additions to the service system, so that in making any cost comparisons on the kilowatt-hour basis, it will be necessary to allocate properly the expenditures for other purposes than the steam-electric plant itself.

In the Installation of new equipment to be operated under vacuum, none of the parts likely to be subjected to vacuum should be painted before erection. The painting should be left until the equipment is entirely erected and started up, with vacuum inside. When equipment is painted while actually subjected to vacuum on the inside, any pores or cracks are thoroughly filled and a visible improvement in the vacuum will be observed. If the painting is done beforehand, the small openings will be covered over by a thin coat and will never become thoroughly filled up.

Regulation of High-Speed Hydraulic Turbines

By JOHN S. CARPENTER
Hydraulic Engineer, S. Morgan Smith Company, York, Pa.

HOW the operating characteristics of the high-speed turbine differ from those of the moderate-speed machine and how this affects the speed regulation of the unit. How to calculate the speed regulation.

IN GENERAL, with speed-regulation problems, we are directly concerned with the amount of power demanded or rejected, the time in which the load change takes place, the flywheel effect of the rotating masses and their speed of rotation. These are basic quantities. There are, however, certain characteristics of hydraulic-turbine runners that exert marked influence on the resulting amount of speed variation above or below normal, and these determining factors can be taken account of in the quantity known as the type characteristic or specific speed. Briefly, the type characteristic or specific speed is the revolutions per minute that a given runner would have, under one foot head, when built of such dimensions as to deliver one horsepower at full-gate opening.

The older formulas for the computation of hydraulic turbine speed regulation were based on a uniform reduction or increase of load within the time of load change, which was approximately true for runners of moderate specific speeds then in use. These turbines,

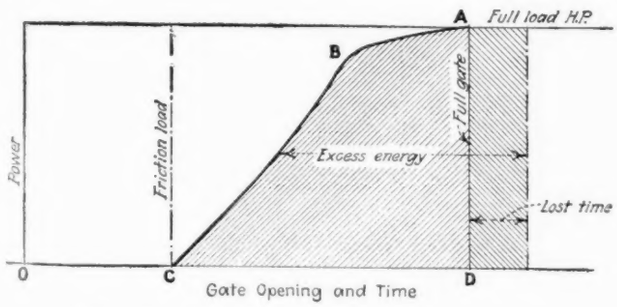


Fig. 1—Shaded area shows excess energy for full load off the turbine

in starting, came up to normal speed at from 8 to 12 per cent of full-gate opening. The power-gate opening curves showed an approximately uniform increase of power up to what is called the "regulating point" of the turbine output. This point is that load beyond which an increase of gate opening is not followed by an increase in power output. Practically, it is desirable to stay somewhat below the absolute power peak, to provide some margin for operation.

In the case of modern high-speed runners, it is found that the higher the specific speed, the greater the amount of gate opening required to bring the turbine up to normal speed, as compared with older types.

For example, a reaction-type runner having its best efficiency around 110 specific speed (English units), requires about 33 per cent of full-gate opening to bring it to normal speed. The Holyoke size best efficiency of this runner is over 91 per cent. A 5-blade propeller having a Holyoke size efficiency of over 92 per

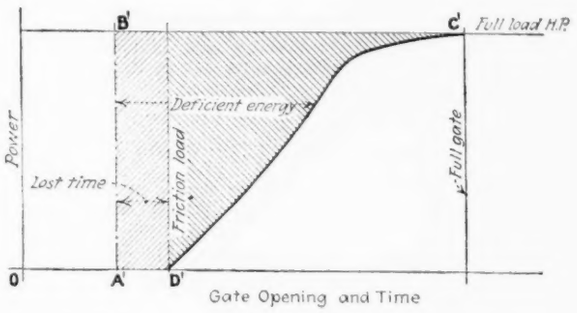


Fig. 2—Shaded area shows deficient energy for full load on the turbine

cent at a specific speed of about 140, attains normal speed at 35 per cent gate. A well-known high-speed runner of best efficiency at 200 specific speed reaches normal speed at nearly 40 per cent gate. It is to be noted that the power curve of the high-speed turbine climbs rapidly and thus, for the case of a turbine running at friction load, full-gate power would be attained in about 60 per cent of the governor stroke. At and near full gate the power increase for a given increase of gate opening is small. Under such a condition, the governor piston would move for at least 10 per cent of its travel from full-load position without affecting the power output appreciably, and what may appear paradoxical, when the load is rejected the output may even increase slightly because the unit power of the runner increases somewhat, in these high-speed turbines, with increase of unit speed.

Regulation With Load off the Turbine

Let Fig. 1 show the gate-opening-power curve for unit head of a typical reaction runner of about 110 specific speed. Assume, for the present, that gate opening is directly proportional to time, then the gate-opening axis can also represent time in proportional units. Assume full load is taken off this unit. The servomotors will not start to move at the same moment that the load is rejected, as the speed must first increase enough to start the governor flyballs moving, and this motion must be communicated to the pilot valve, the pressure oil started flowing, etc. This time delay may take from 0.1 to 0.3 second, depending on the governor design and its inherent details. This interval is "lost time" and the speed keeps on increasing meanwhile. Supposing that there is no pressure rise or drop in the penstocks, the energy affected is that inclosed in the

area A, B, C and D, to which must be added that of full output for the "lost time." The mean power during the regulating time is the energy that goes into the revolving masses and increases the speed of the turbine.

The foregoing is true with modifications. For example, the rate at which gate opening is changed is not a direct proportional to time; as the servomotors start to move slowly at first, increasing in velocity until the pistons are quite near the end of the stroke, where the ports that carry the oil discharged from the cylinder are reduced in size and consequently slow down the piston speed. When starting up the turbine, the water quantity demand at no load will be such that a 30 to 35 per cent gate opening will be required, depending on the specific speed of the runner. When closing the gates due to a load rejection, the effect of penstock pressure increases and the energy stored in the rotating mass being given out, will cause power to be developed in excess of the no-load requirements when the gates are closed beyond the no-load position.

Regulation With Load on the Turbine

Allowing this case to rest here for the present, let us take up the "load-on" condition, Fig. 2. Starting with the unit running at friction load, suppose that full load is applied. In this case there is also a lost-time interval before the gates start opening. Assuming that the load came on suddenly, as by closing a switch, there must be supplied from the kinetic energy contained in the rotating masses, the energy represented by the area A', B', C' and D' for the time required to open the gates, since the turbine supplies the balance of the power. From a comparison of the shaded area in Figs. 1 and 2 it can be seen that the amount of deficient energy, in case of load on is less than the excess of energy to be absorbed in the case of load off. This is due to the shape of the power curve ABC. If this curve were a straight line between A and C, and the other conditions in the hydraulic and mechanical system remained the same in both cases, then the amount of deficient energy for load on would equal the excess energy for the same amount of load off. For the conditions shown in Figs. 1 and 2, when the head does not change, the unit inertia being of normal values, and the gate opening and closing times are equal, the resultant speed drop for load on will be less than the speed rise for load off.

By reference to Holyoke tests of modern high-speed wheels it will be found that the power varies appreciably with the speed at one foot head. Therefore, when computing the mean excess or deficiency of energy over the regulating time it is necessary to correct for this variation in output at several points on the gate opening-power curve. Also, the effect of non-uniform velocity of the servomotor piston must be allowed for. In the computations of the coefficients or form factors the last-named correction was made by comparisons with tachograph test data at hand, while the former was checked by step-by-step computation with Holyoke and laboratory tests.

Effect of Pressure Rise and Drop

The writer computed a number of cases of regulation with pressure rise and drop; using step-by-step numerical methods. It was found that this effect on resultant speed variations is taken into account by using as a factor the square root of 1 plus the rise or drop as later indicated. This conclusion was later checked against a tachograph test of a modern plant in New York State, where the pressure rise for full load off is over 110 per cent, normal head being about 145 ft.

The rise in speed, N in revolutions per minute, caused by an average excess of power, $Hp.$, over a regulating time, t seconds, with a given amount of flywheel effect WR^2 is given by the formula:

$$N^2_2 - N^2_1 = \frac{3{,}230{,}000 \times Hp. \times t}{WR^2}$$

If the turbine output is stated in terms of its full load horsepower and a form factor F is used to correct the same to average power, and at the same time express the pressure variation, up or down, as a decimal k, the final formula becomes

$$N^2_2 - N^2_1 = \frac{3{,}230{,}000 \times Hp. \times t \times F\sqrt{1+k}}{WR^2}$$

N_2 being the final speed in revolutions per minute and N_1 the normal speed. In the case of load on, the subscripts of N are reversed, as the final speed is less than the normal value.

A table of form factors F for load on and off follows, n_s being the specific speed in English units:

VALUES OF FORM FACTOR F

n_s	Full Load Off	Full Load On	n_s	Full Load Off	Full Load On
20	0.56	0.45	120	0.66	0.37
40	0.58	0.43	140	0.68	0.36
60	0.60	0.41	160	0.69	0.35
80	0.62	0.39	180	0.70	0.34
100	0.64	0.38	200	0.71	0.33

It should be noted that different designs of wheels will have values of F slightly different when compared with others of the same specific speed, although all the wheels compared may have very good efficiency. It merely means that some wheels have better regulating qualities than others, all other factors being equal. Some types of propeller turbines have so much flywheel effect in the runner itself that it is worth adding to the generator flywheel effect.

Load-On Conditions in Open Flumes

When figuring load-on conditions in open-flume types of settings having fairly long canals laid on a slight grade, it should be remembered that it is easy to draw down 10 to 20 per cent of the head by a sudden load demand. The problem is to get the water to the turbine quickly enough for load-on cases and to dispose of it by spillway or surge tank as quickly in the case of load off.

From test data at hand the speed changes due to fractional load changes, stated in per cent of the full-load change, are as follows, and are good average values:

Per Cent Load Change	Per Cent of Full Load Speed Change	Per Cent Load Change	Per Cent of Full Load Speed Change
100	100	25	15
75	60	10	6
50	35		

The matter of pressure rise and drop does not appear to affect the relative proportions given in the foregoing, which was checked out in the case of the previously mentioned plant with the abnormal pressure rise. It is suggested that those interested check up the foregoing formula with the published tests on governors at the Holtwood plant of the Pennsylvania Power & Water Co., which was published in *Power* Sept. 18, 1923, and Oct. 9, 1923, issues, remembering that the opening and closing times of the governor are not equal.

Reheating

IN POWER stations employing steam pressures of five hundred pounds or over, the reheating cycle becomes practically a necessity. There are now four or five stations in this country and one in England operating with such a cycle, and as the use of high steam pressures seems assured of further extension, reheating is likely to receive continued attention.

In each of these present stations the boiler type of reheater is used and in spite of certain complexities is reported to be operating successfully. An alternative arrangement is that of reheating with live steam, as advocated by Mr. Broido in an article in this issue. In its favor may be urged lower initial costs and greater simplicity of control. Offsetting these points is the fact that with such an arrangement the steam cannot be reheated to the same total temperature as the live superheated steam—not within twenty-five or thirty degrees of that temperature. But Mr. Broido contends that this is unnecessary; that by reheating further down the expansion curve the steam can be kept dry throughout the turbine and that this is all that is necessary.

Admittedly, the gain in economy due to reheating to the higher temperature is somewhat greater than to the lower temperature, from a B.t.u. standpoint. Each of the stations employing reheating has shown remarkably good performance, to which reheating has undoubtedly contributed, but to what extent it is difficult to say. Sufficient data are not yet available from which to determine the net gain due to reheating when balanced against fixed charges on the initial investment. The final answer must be measured in terms of reduced cost per kilowatt-hour, all factors considered, from the standpoint of fixed charges and operating considerations. It is upon this basis alone that the merits of the two systems should be judged.

Pulverized Coal and Stoker Problems

IT IS well to face our difficulties, and there is no denying the fact that any departure from long-established practice is bound to develop difficulties. Pulverized coal is no exception. Used for years in the cement industry, its adaptation to the boiler furnace was hailed by some as the panacea for all the ills of the boiler room, but experience has confirmed the view of more sober minds that the remedy has brought with it incidental effects that leave room for improvement.

This is in no sense a reflection on the ability of those who make or use pulverized coal equipment. Every forward step in the art of power generation has unexpected turns. The stoker is a splendid example of this situation. Through the last twenty years its history has been one of problems faced squarely, attacked intelligently, and solved more or less completely. The present stoker is strikingly different from its forerunners.

In this issue there are two articles, one stating some of the problems confronting the designer and user of pulverized-coal equipment, the other describing recent developments in the underfeed stoker field. With the ability and energy now being devoted to the development of pulverized coal, it is safe to expect that continuous improvement will mark the use of pulverized coal just as in the case of the stoker.

The important thing is that no prudent designer, whether of a new thing or an old one, dares claim perfection, but rather proclaims his problems and devotes his energies to lightening the burden of the plant operator through their solution.

The Paradox of Power-Plant Investment

WHILE the general method of finding the most economical thickness of heat insulation has been known for some time, the necessary data have not been available in a form suited to rapid computation. Nor are underlying principles as widely understood as they should be. For this reason the article by L. E. Whitaker on page 762 of this issue is a timely contribution to the literature of the subject. Mr. Whitaker clearly outlines the fundamental principles of "economic thickness" and makes them usable by a series of total-heat-loss charts for various combinations and thicknesses of insulating materials.

At first glance it seems paradoxical that the thickness of insulation which yields the highest percentage return on the investment is rarely the one most economical to use. The thinnest commercial thickness of covering invariably yields the highest percentage return. The reason for this is that the cost of the job increases in approximate proportion to the thickness, while the heat loss falls rapidly at first and then more and more slowly. Doubling the thickness practically doubles the cost, but cannot by any possibility cut the loss in half. The thinnest covering available will save about three-quarters of the heat that would be wasted from a bare pipe. Piling the thickness up to infinity could do no more than save the other quarter.

As pointed out in the article on instruments (page 766) and neatly illustrated by Mr. Whitaker's article, money-saving appliances differ from ordinary investments in stocks and bonds in the limitation put on the amount of money that can be invested at a given rate of interest. The thinnest covering available may easily yield a return of two hundred per cent or more on the investment, but the amount that can be invested at this astonishing rate is limited to the cost of covering the existing heat-wasting surfaces with such a layer. Having exhausted the possibilities of making two hundred

per cent investments, any good business man would be interested in a chance to earn fifty per cent, say, on the additional cost of the next greater thickness.

The proper procedure in all such cases is to consider a series of propositions of progressively greater cost and pass from one to the next as long as each increment in cost yields an attractive return.

The Passing of the Horsepower

THE automobile is crowding the horse off our city streets and country highways, and up-to-date rational methods of technical thought are crowding the horsepower from our engineering language.

The boiler horsepower is well on its way to oblivion. The physical size of a boiler as a structure is more and more being stated in square feet of heating surface, and the engineering profession is gradually learning to express the load on a boiler in terms of the rate of heat absorption for the boiler as a whole and the rating or relative load in terms of the rate of heat absorption per square foot per hour.

Recently, another horse has been started on his way to the happy hunting grounds. At a recent meeting, the Advisory Committee on Prime Movers of the International Electrotechnical Commission went on record in favor of suppressing the use of horsepower for the expression of turbine output, and the substitution of the kilowatt in its place.

Attention was directed to the fact that the kilowatt is not an electrical unit, but a strictly mechanical unit, in precisely the same way that the horsepower is a mechanical unit. The horsepower is defined as 33,000 foot-pounds per minute, or 550 foot-pounds per second, while the kilowatt is defined as 10^{10} dyne-centimeters per second. The dyne is a perfectly definite force, being defined as the force which will impart unit acceleration (one centimeter per second per second) to unit mass (one gram). It is true that these units are not in common every-day use in this country, outside of scientific laboratories, where they are almost universally used the world over, but nevertheless they are strictly mechanical units. The dyne is a force just as the pound is understood to be in ordinary parlance. The centimeter is a distance exactly as the foot is. The dyne-centimeter is a unit of work, being called an erg, exactly as the foot-pound is a unit of work, although the latter has no special name.

The watt and kilowatt are useful in measuring electrical energy only because the basic electric units, the ampere, the ohm and the volt, are defined in terms of the dyne and the centimeter. Had the electrical units been established in other terms, the horsepower might have been the appropriate unit.

Those who have not been accustomed to working with units of the metric system will, of course, oppose a change of this nature, on the ground that it complicates the calculation of results. On the other hand, it must be pointed out that it is just as easy to calculate power in kilowatts as in horsepower even in the English system of units, and in the metric system, used by many of our fellow engineers in other parts of the world, it is much easier. For example, in calculating the indicated horsepower of a reciprocating engine, the area of the indicator card is reduced by the application of scale constants and engine speed to give foot-pounds per unit time, and division by either 550 or 33,000, depending on whether the time is one second or one minute, gives the horsepower. This calculation will yield kilowatts if the divisor be changed from 550 or 33,000 to 737 or 44,200, respectively. With the slide rule in almost universal use, it is just as easy to divide by one number as by another, and the only objection to the new quantities is that perhaps one will not remember them as readily as the old ones. This is a small matter because experience shows that even the most awkward numbers are readily remembered by those who have occasion to use them now and then. For example, few have difficulty in remembering the value of π, and most engineers could write down from memory a surprisingly long list of numerical constants affecting their work.

Of course, the horsepower will continue in use for a long time, but it is good to know that the present confusing set of units is gradually being simplified by the elimination one at a time of units having the most objectionable features. The transition from an irrational confusion of numerous systems to a single rational system is an undertaking to which the engineering profession must seriously address itself, recognizing, however, that such changes can be made only slowly, one step at a time.

Cooling Water Systems for Oil Engines

CONCEDEDLY, much of what difficulty there is experienced in oil-engine plants is due to the inferiority of the water-cooling systems. When the available raw water carries little or no scale-forming material, its use directly in the cylinder jackets is economically advisable. Unfortunately, most of the oil-engine plants are located in districts where the water is entirely too bad to be introduced into the jacket or piston. In the face of this it is by no means unusual to find such water being used, and if a cylinder head fractures, the cause is easily shifted to mysterious heat stresses.

Three parties are deeply interested in this problem —the engine builder, the owner and the plant engineer. The builder appreciates the necessity of a proper water supply, but unfortunately too many sales engineers, after obtaining the contract for the generating unit, are disinclined to point out to the purchaser the necessity of an additional expenditure in constructing a well-designed cooling system. The purchaser is at best not deeply versed in engineering and, even if he were inclined to be generous in the capital investment of the new plant, does not understand the vital need of non-scaling water. Obviously, the operating engineer must, so far as possible, insist upon the use of the proper equipment.

The most satisfactory cooling arrangement is undoubtedly the closed system in which only rain or distilled water circulates, the heat picked up in the engine jackets being finally removed by a stream of raw water flowing over a set of coils through which the raw water is pumped. The investment is not high, and the sole operating expense is the extra power required by the greater pumping head.

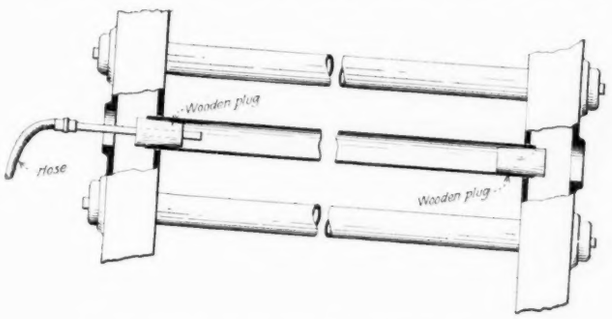
If a closed system is impossible, chemical treatment of the raw water introduced into the system should be adopted. The cost may be less than the pumping expense of the closed system, but entails some little attention from the engineer.

Practical Ideas From Practical Men

Home-Made Tube Tester

In our plant of 8 water-tube boilers, during the winter we have more split tubes than at any other season. The majority of these defects occur in tubes of the lower rows nearest the fire, but occasionally one splits somewhere in the midst of the tube nest, and valuable time is lost in locating the tube to be removed.

There have been cases where the wrong tube has been



Plugs and hose locate the split in the tube

cut out; this is not only aggravating but expensive in both time and material.

In order to eliminate any chances of making this mistake and having no manufactured device at hand for this purpose, our foreman used a couple of pieces of wood and scrap pipe from around the shop to construct an apparatus for this purpose.

The pieces of wood were turned down to a proper taper to fit the tubes, being about 12 in. long. A hole was bored all the way through one of them to allow a snug or tight fit for a short piece of pipe, which extends into the hole about half way and is connected to a hose by means of a union. The other end of the suspected tube is fitted with a solid plug of wood.

When the boiler is drained of its water and the device is applied to a tube that is thought to be leaking, little time is lost in locating the right tube since the water will be forced out through the split.

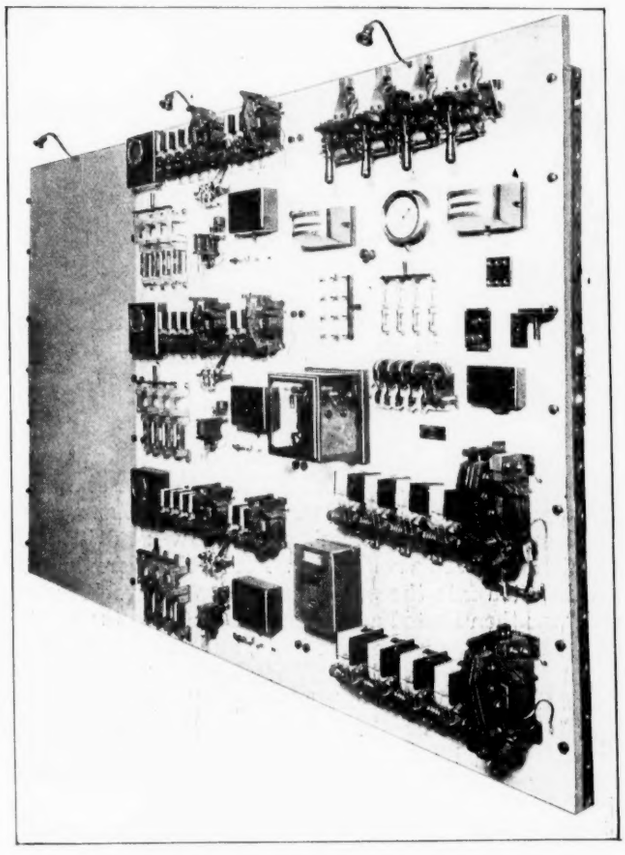
Pittsburgh, Pa. J. W. GRUNDY.

Unusual Switchboard for Pump-Motor Control

The panel shown in the figure was built to control three 40-hp. induction motors driving pumps in the Philadelphia sewage system, and to provide for switching of two banks of transformers that supply power for the motors.

Pumping is controlled automatically by means of pressure switches governing the number of motors in use. The power for running the three motors comes from two banks of transformers, one 50 kva. and the other 100 kva. By means of interlocks on the line contactors of each of the automatic starters, the 50-kva. transformer bank is connected to the panel whenever one motor only is running. When two motors are in use, the 50-kva. bank is disconnected and the 100-kva. bank is switched in. When all three motors have been installed and are running, both banks of transformers will be used.

The panel bases are of polished white Italian marble, each measuring 4 ft. wide by 8 ft. high. Two panels are now in use and a third is provided for future expansion. Among the devices mounted on the panels are three separate automatic resistance-type starters with current-limit acceleration, pressure switches for control, and thermal relays for overload protection. After com-



Front of panels for automatic control of three motors and two banks of transformers

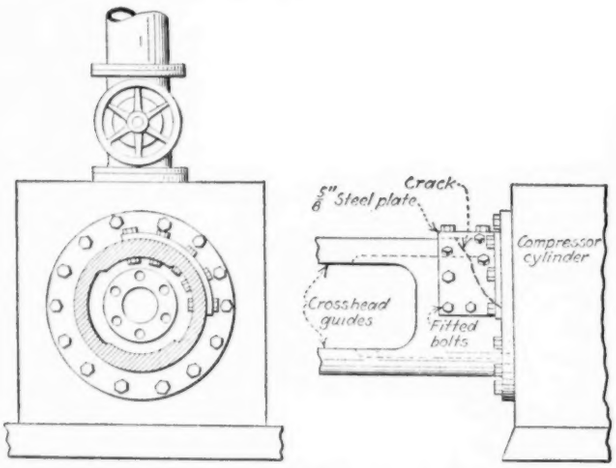
ing through the transformer contactors the power goes through a four-pole air circuit breaker, which has an overload trip in each of the four circuits. In addition to the circuit breaker and transformer contactors, the panel has a voltmeter, an ammeter, a curve-drawing wattmeter, a watt-hour meter, a pressure gage and an automatic starter for a 5-hp. motor.

Schenectady, N. Y. B. S. HAVENS, General Electric Co.

Repair to Ammonia Compressor

A crack developed in the guide trunk of a heavy-duty compressor close up to where it joins the cylinder starting on top and traveling around the side till it ended at one of the bolt holes in the cylinder flange.

As it was not convenient to stop the compressor at the time, a heavy iron strap was run around the guide to hold the crack in check until the machine could be



Repair patch fitted over crack

taken off the line. On examination it was found that the cylinder had been lined up with metal wedges, and shims, which had worked out, throwing the weight of the cylinder on the guide trunk.

The first thing we did was to line up the cylinder, with the face of the guides, and drive the wedges into place and hold them there with fresh grout, the clamp being used to hold the crack from spreading during this operation. The clamp was later removed and a hole drilled at the end of the crack to prevent further spreading.

It was then decided to make a patch and hold the two pieces in place. To do this a tin templet was first made, then a piece of ⅜-in. plate was heated and hammered into shape. A sufficient number of holes were drilled in the patch to insure a solid joint. Four holes were drilled off this patch into the guide straddling the crack. The patch was then heated and laid in its place on the guide and made fast with the four bolts, and then hammered down until it was a good fit. When it had cooled, the rest of the holes were drilled through the guide trunk, using the patch as a templet. These holes were then reamed out and fitted bolts put in. The bolts were driven from the inside and the nuts set up as tight as possible with a wrench and hammer, making a repair that was good for the life of the engine.

Hollywood, Calif. MAURICE C. COCKSHOTT.

Information on Blueprints

The manufacturers and distributors of power-plant equipment and specialties give too little information on their blueprints and as a result much time is wasted in erection.

Informing a customer of sizes of openings for pipes, also how valves are piped, whether to high-pressure side, low-pressure side, vacuum or atmosphere, would be appreciated.

Would it not be advisable for all manufacturers to adopt a standard practice of putting the weight of the apparatus they furnish on the blueprints they submit to the customer?

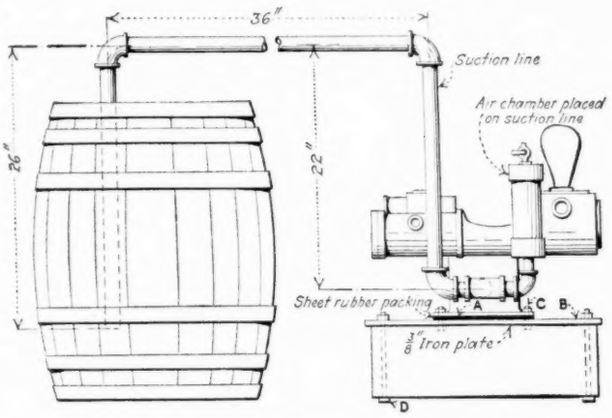
It is surprising to see how much valuable time is lost in a large organization due to this lack of necessary information. LOUIS R. ENGEL.

New York City.

How Slamming of Pump Was Overcome

On attempting to start the small boiler-feed pump, shown in the accompanying illustration, after the plant had been closed down for two years, it refused to draw the water from the barrel. It will be noted from the sketch that the suction line enters the base of the pump. Upon removal of the pump from its foundations it was discovered that in closing down the plant the draining of the pump base had been overlooked. Consequently the water remaining in the base froze and pushed the bottom of the base out, putting the pump out of commission. As it was necessary to place the pump in operation as soon as possible, a piece of sheet-rubber packing *A* was cut to fit the pump base and a piece of ⅜x8x28-in. iron *B* was drilled to take the regular foundation bolts. The rubber packing was next placed on the pump base, covering the break, and the iron plate was bolted over it, using the short bolts *C* in the regular foundation bolt holes to pull the joint tight. The pump was then bolted to its foundation, which was of wood with other bolts *D*. When the pump was again started, it operated satisfactorily.

From the time this pump was first started it slammed with sufficient violence to cause excessive vibration,



Air chamber on suction line stopped pump slamming

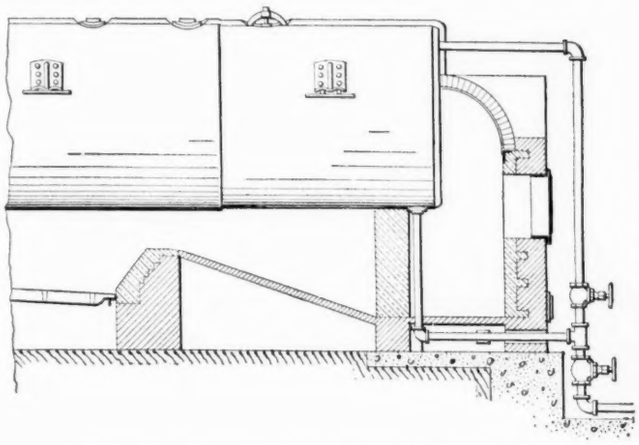
and it was concluded that the trouble was caused by the momentum of the column of water in the suction line at the instant the pump reversed. An air chamber was installed on the suction line as indicated on the sketch. This partly remedied the trouble. A pet-cock was next connected in the top of the suction chamber and adjusted while the pump was operating. This practically eliminated the troublesome slam. The adjustment of the pet-cock is changed slightly when the water is high in the barrel. The slam could have been eliminated by changing the suction piping, but this was not desirable owing to other conditions.

Wise River, Mont. F. W. GOLDSMITH.

Comments from Readers

Protecting Blowoff Lines

In the Jan. 5 issue of *Power* F. E. Bateman asks for a plan to protect blowoff pipes. In the illustration is shown a simple method that I have used for a number of years. Connect a pipe from the steam space outside of the rear wall down to the blowoff pipe, with a valve



Method of preventing blowoff pipe trouble

to close when blowing down. One then has a constant circulation through the blowoff pipe and blowoff troubles are over. There is no need of covering the blowoff pipe.

W. A. FISHER.

Exhaust vs. Live Steam for Heating Purposes

I wish to comment on some principles in John Gr. Smedley's statement on "Exhaust vs. Live Steam for Heating Purposes," in *Power* of March 16.

We have a non-condensing turbine belted to compressors through a lineshaft. During the heating season a back pressure of 30 lb. is maintained to heat two large mills and an office with the exhaust. A fan forces air through the heaters, and the warm air is piped to various positions in the mill, the heaters being situated centrally in each mill and being drained by traps. The heaters are approximately 300 and 200 ft. from the source of steam supply.

In the first installation the steam was transferred through a 3-in. line; this being unsatisfactory, a 10-in. line was installed, with better results. But on the 300-lb. transfer the steam condensed so badly that the system was not practical, although 20 lb. was maintained on the heater. Live steam was the only remedy for the office and the large mill, it being sent to the heater at 510 deg. total temperature, but at the same pressure as the exhaust steam. If the steam was maintained at the same pressure throughout, with equal amount of velocity at all points, live steam would give better results because it contains more heat units to be distributed, and there will be less flow of steam per hour. At a rapid velocity the steam does not have a chance to radiate in its passage of a given point or points.

In a steam boiler the fire might be burning at a rapid rate and have a high stack temperature, yet you might be getting poor efficiency and low CO_2.

S. W. HAY.

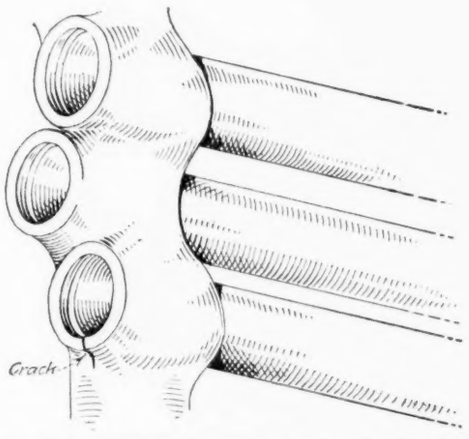
Bedford, Ind.

Hydrostatic Test for Boilers

I have been interested in the recent discussion on the hydrostatic test for boilers and I should like to add to what has already appeared on this subject. In my opinion there are times when a hydrostatic test is a good thing and should be used, and I will cite one instance to show where it did good.

On a large horizontal water-tube boiler, with cast-iron vertical headers, that had been shut down for cleaning, a thin coating of scale was found on one of the headers near the bottom. The scale was scraped off but the workmen reported that no sign of a crack could be found. When the cleaning and repairs were finished, the boiler was put back in operation for another period of thirty days, which was the length of time between cleanings.

On the next cleaning another lot of scale had formed in the same place, but thicker than before. The scale



Crack in header showed up when hydrostatic test was applied

was cleaned off and the boiler closed up, then a 150-lb. working pressure of the boiler hydrostatic test was applied, which showed the header to be cracked at the bottom as illustrated. The contraction caused by the cold water, opened up the crack so that the water could be seen coming out. A crack like this might have caused a serious accident if it had not been found. The eye of an experienced man and a small test hammer could not have located this leak.

Norway, Mich. THOMAS PASCOE.

Another Ammonia Fallacy

While dispelling some ammonia condenser fallacies in the March 30 issue, H. J. Macintire says: "If the action is such that diffusion, which varies as the square root of the vapor density, is not possible, as is usually the case in pipe condensers during operation, and the air collects at one end of the condenser, it exerts an action similar to that of a piston on the entire condenser and liquifaction will take place at a temperature corresponding to the total pressure $p_1 + p_2$, even in the zone which has practically pure ammonia."

I wonder if Mr. Macintire has ever tried to visualize the action he describes; that is, just how the air "collects" at one end of the condenser? Under what conditions is diffusion rendered impossible? How can an accumulation of air in a condenser act like a piston?

Los Angeles, Calif. C. O. SANDSTROM.

Babbitting Bearings

The letter from E. Darwell, on page 501, March 30 issue of *Power*, contains some excellent advice. He suggests using putty instead of clay for closing the ends of bearing shells while rebabbitting them.

This is usually the material used when rebabbitting all small bearings, clay being seldom if ever used. When working large bearings the case is different. Most men who have used putty alone in the latter case will perhaps recall times when the run was spoiled because the heated putty at the bottom or side of the bearing moved slightly, because of the pressure of the molten metal, and allowed some of the metal to flow out. The danger of this happening can be greatly lessened if a little oil and heat are used to soften the putty and it is then rethickened and restiffened by working in powdered asbestos or mineral wool.

When this is done, it will require considerable more heat and pressure to move it than is the case with putty alone. Putty treated in this way has been used by water and gas main pipe layers for years, for coating the rope runners instead of using clay. The putty mixture does not require resoftening after every few runs, as does clay. Of course the factory-made asbestos joint runner, or dam, with clamp attached, is generally used with running pipe joints, but there are many men who stick to the old treated rope method.

Toronto, Canada. JAMES E. NOBLE.

What Is the Engineer's Standing

Carl F. Miller in *Power*, March 23, has given a very good reason why he has discontinued taking the journal. He states that he is receiving eight cents more per hour for doing work that does not require any special training, or, to speak more plainly, for doing work that any unskilled laborer can do.

When he dropped out of the engineering profession he did just what hundreds of other engineers have done and for the same reasons—that is, long hours, low pay and lack of appreciation on the part of his employers as to the engineer's efforts to get the best results from the equipment under his charge. As a rule, the engineer's position is a responsible one. Steam, mechanical and electrical equipment, the value of which runs up into thousands of dollars, is directly under his charge. He is held responsible by either the government, state or city authorities and insurance companies, sometimes by all four, for the safety, health and comfort not only of those within the building where he is employed but in a good many instances of those on the outside as well. To be proficient in his calling he must be master of many different trades or professions. Only by years of increasing toil and study can he obtain the necessary experience properly to hold an engineer's job so that he may give his employer the service that he should get.

In the first place, he must know all about boilers, engines, turbines and pumps. Secondly, he must be an electrician, carpenter, pipe fitter, plumber, brick layer, steel worker, machinist and a lot of other things too numerous to mention.

He must have a library on engineering subjects as well as subscribe to one or more technical papers and magazines, for if he stops studying he soon becomes out of date. For all of this he receives less pay than any one of the above-mentioned workers.

The Foreword in the March 23 issue, addressed to the employers of engineers, is timely. It is hoped that some of them will take time to read it and think it over. They should be brought to a realization of the fact that the engineer is a very important cog in the wheel of business. Very often the power plant and the man that operates it are determining factors as to whether his employer's business shows a profit or a loss at the end of the year. It is not fair to engineers that they should be compelled to seek other means of livelihood where working conditions are more attractive and the compensation is more, but no doubt that they are justified in acting the way they did, for eventually they will better themselves financially, as well as have better health and enjoy more peace of mind. JOHN H. SCHMIDT.

Baltimore, Md.

Purchasing Coal on Specification

Otto Kohr's comment in the April 20 issue on H. D. Fisher's letter about my article, "The Purchase of Coal on Specification," is welcome because it sets forth the view which I have been hesitant about expressing, namely, that Mr. Fisher has missed entirely the main point of the article. I feel that the views presented by Mr. Fisher are the result of insufficient knowledge of mining technology and the use of coal, and not intentionally quibbling.

The original theme of the article, "that by experiment, the stokers available should be adapted to the coal available, and then some means taken to protect against carelessness nullifying all the previous experimental work, by laying down purchase specifications," must be admitted by those who have gone through the stoker experimental stage, to be incontrovertible.

The Engineering Experiment Station of Pennsylvania State College, in its Bulletin No. 14, presents results secured from inquiries to a few of the larger coal buyers, showing that in these cases there is no quibbling or see-sawing between buyer and seller over methods of sampling, and it is common knowledge that the sampling problem is the smallest of the many connected with the purchase and use of coal, among all the larger buyers of coal in this country.

Therefore is it not likely that the point discussed by Mr. Fisher was of such small moment as to have been better left untouched? CHARLES H. COLBORN.

Jeannette, Pa.

How About the Fireman's Pay?

In the March 23 issue, the editor has written an excellent Foreword concerning "Power-Plant Engineers' Wages." These wages will eventually be received by some who are now firemen.

Let us ask another question: "Are stationary firemen paid their worth?"

Numerous power-plant executives will admit that the boiler room is the heart of the power plant, but this department is receiving the least consideration of all, especially concerning wages. Modern appliances for the saving of time, labor and money may be installed. Still, in this era of advancement, man has not yet been entirely replaced by machinery. Proper firing can not, or has not yet been accomplished without the aid of man.

An intelligent fireman can be very active in the savings produced by the management of a power plant, if he is so inclined. However, if he is not amply compensated for his endeavors to effect economy for his employers and with the vision of the present rate of engineers' wages before him, how can he be expected to remain in the game?

It may be possible that Carl F. Miller is right in his letter which was made the basis of the editorial in question.

More power to the engineer, but a little more consideration in general should be shown toward the firemen in their present positions. JAMES W. GRUNDY.

Pittsburgh, Pa.

Engineering Students Dropping Out

In a communication appearing in the March 16 issue E. Ogur states that 80 per cent of the engineering students in our colleges enroll as a result of deliberate misunderstanding of the opportunities that engineering offers, and further suggests that the lack of economic, spiritual and financial independence experienced by engineers is such that few would have taken up this line of work if they had been aware of the conditions confronting them. After reading his letter one is led to the conclusion that, in his opinion, engineering is not so much a profession as a mistake.

Now if he means that the majority of engineering students have been deliberately misled by some person or persons unknown, I cannot agree with him, but if he means that 80 per cent of the college freshmen entered college with very little knowledge of the opportunities, or lack of opportunities which exist in the field for which they had decided to prepare themselves, I most heartily agree with him. I make this statement as the result of many intimate talks that I have had with recent graduates, but in spite of this it does not follow that most of these young men have made a mistake in becoming engineers; in fact, I am quite convinced that most of them have not.

During the years in which I have been connected with engineering, I have doubtless become acquainted with several thousand men engaged in this line of work, and although most of us at times have envied those engaged in more remunerative lines, and sometimes even decided that we would ourselves try something else, we have generally, I think, felt that it was not money that was binding us to our chosen field, but a more fundamental urge which we would ignore at our peril.

It is a most obvious thing to assume that the governing factor in deciding upon one's life work is the question of remuneration, but like most other apparently obvious things, this is far from being generally true, and in connection with this we may learn a lesson from history.

What art was to Greece, what the conquest of the sea was to England and other great maritime nations of the past, so are the conquest and utilization of the forces of nature to those who have inbibed the spirit of America.

England has never lacked men for her navies; sons of her most famous families have forsaken surroundings of ease and luxury to live under conditions which, until within recent years, did not afford the comforts enjoyed by the average American laborer of today, and seldom did the monetary return cover the cost of living. Many a merchant seaman has advised his sons not to go to sea, but as for himself, what could he have done but follow the call of his heritage, in spite of all hardships and dangers, and today in this country we have thousands of young men who are being urged by an overpowering racial instinct to enter the engineering field.

Fortunately, the community can use them all, and could use many thousands more if only the possibilities were more clearly recognized.

This is a mechanical age, and America stands head and shoulders above all other nations in the use of machinery and engineering processes. Nearly everything we touch, eat or drink, is an engineering product. A woman may sew by hand, but it is with a machine-made needle, and machine-made thread, on a machine-made fabric. She may cook, but only with machine-made utensils on a machine-made range. The very water we drink comes from machine-dug reservoirs or machine-driven wells, and is pumped by machinery through machine-made piping. Indeed we might almost ask in these times, Who should not have an engineering education?

If we would only spend a small part of the money and time in educating the public, not forgetting ourselves, as to the nature of the services that technically trained men can render to the community, which is spent by some of our more progressive friends in creating a market for their products, I think that people would not have so much cause to lament the lack of economic, spiritual and financial independence among engineers. I admit that under existing conditions there is a certain amount of reason for the complaint, but the remedy is to a large extent in our own hands, and the problem will not be solved by refusing to carry on, and conquer our difficulties.

This is a new age, but too many of us are thinking in terms of an age that is past. The fight today is not for political justice but for economic justice, and no country has made greater advances in this direction than our own; semi-pauperism tempered by charity, has no legitimate place in our modern economic philosophy, the need for services of any kind at less than their legitimate value does not exist. The first commandment of industrial ethics is, "Remember that the laborer is worthy of his hire."

Our particular duty as engineers is to keep this law, and to see to it that our young men have every opportunity to make themselves of the greatest possible value to the community, and that the community pays for what it gets. JAMES O. G. GIBBONS,

Newark, N. J. Consulting Engineer.

Questions and Answers

Oil in Boilers

Will a slight amount of cylinder oil reaching the tubes of a boiler cause them to bag or blister?

If any appreciable amount of oil goes on the tubes or shell, bagging or blistering may occur. This is for the reason that oil is an excellent non-conductor and interposed between the metal and the water, prevents the water absorbing the heat passing from the fire to the metal. As a result the tube overheats and may bag. Oil should be kept out of a boiler.

Troubles with Superheated Steam

What are the disadvantages of using superheated steam?

Operating engineers found difficulty in keeping gaskets tight when first using superheated steam, and some valve and pipe fittings were not suited to it. These troubles have been overcome, and aside from that, there is no disadvantage in the use of superheated steam. For many purposes the use of superheated steam results in marked economy in fuel. This is notably true where steam turbines are used.

Will Liquid Damage Ammonia Compressor?

If the expansion or shut-off valves are at top of the wall coils of refrigerating plant, can the gas come back to the machine wet enough to cause damage to the compressor cylinder?

If you are carrying the coils completely filled with liquid, there is danger of slopping over of the liquid into the suction line upon starting, provided the coils are filled with liquid and the expansion cocks have not been shut, and the compressor is pumped down somewhat before closing down.

Finding Horsepower of Centrifugal Pump

Kindly explain the method used in calculating the power needed to operate the conventional type of centrifugal pump. Explain in a brief way, taking a certain size pump, figuring on a given lift and head. I have a 25-hp. high-speed gas engine driving a pump and I wish to figure the load on it.

Assume a 5-in. pump delivers 600 gal. per min. and that a pressure gage placed close to the discharge connection to the pump registers 21 lb. per sq.in., and a vacuum gage placed at the suction elbow of the pump shows a vacuum of 10 lb. The vertical distance between the two gages will be taken as 2 ft. The work done is the lifting of the weight of water, or 600×8.33 lb., the vertical distance plus the friction losses (included in the gage reading). This is equivalent to 2.33 $(21 + 10)$ plus the vertical distance between the gages, or 2 ft. Work $= 600 \times 8.33\ (2.33 \times 31 + 2) = 371{,}002$ ft.-lb. or $\frac{371{,}002}{33{,}000} = 11.2 +$ horsepower. The efficiency of the pump would be about 70 per cent, making the horsepower required at the shaft $11.2 \div 0.70 = 16$.

Leaks in Carbon Dioxide Refrigerating System

In operating a CO_2 refrigerating machine we lose heavy charges of gas. Please advise how and by what reagent can the leaks be located around the machine and in the brine tank.

If there is any large amount of carbon dioxide in the brine, the acidity of the resulting mixture will cause blue litmus paper to turn pink. It would be possible, however, for you to have a considerable loss in the brine before the brine would become acid enough to cause action on the paper. To detect leaks at joints, the safest thing to do is to use soapsuds on the suspected spot, and the carbon dioxide blowing out of a leaky joint or gasket will cause bubbles to form in the soap lather. You should have your carbon dioxide treated with peppermint or some other strong-smelling flavoring extract, so that a leak could be detected by the strong odor of the air around the joint.

Boiler Installed in Cramped Space

Two 3,400-ft. cross-drum water-tube boilers are to be installed in a basement the height of which is so restricted as to require special design of the boiler and arrangement of the connections. The steam and water drum is only 42 in. in diameter. The front circulation tubes enter the drum on a level with the center, whereas circulating tubes to rear header leave at a point only 12 in. lower, thus producing a pocket of dead water 9 in. deep in the drum. The steam supply leaves from the top of the drum with a dry-pipe attachment. Will there be any pronounced tendency to surge? The head room will not permit the main steam valve to be set in a vertical position. Is it practicable to use stop and automatic-stop valves with the stems horizontal? Rotating-head soot blowers are to be installed. Is it the custom to install dusting doors in the settings, aside from the openings for these blowers, or can they be eliminated?

With a drum of restricted diameter and such a pocket of dead water there would be danger of surging. If the manufacturers have not already taken care of this, provision should be made for draining the dead water from the steam drum back into the circulating system of the boiler.

There is nothing to prevent using ordinary stop valves with the stems in a horizontal position, although some inconvenience may be expected from leakage of condensation around the valve stems. Perhaps space will permit tilting the valve stems slightly above the horizontal position to keep condensate from the stuffing box. Metallic packing will help. As to automatic stop valves, they should be installed if it is possible to manage it. It is possible to obtain such valves designed to operate in a horizontal position. Before proceeding with the erection it would be well to get the approval of the local boiler inspector.

The installation of dusting doors in the settings may be a convenience at times, but they are likely to cause losses due to radiation and air leakage.

News in the Field of Power

National Electric Light Men In Convention at Atlantic City Plan Industry's Progress

Thomas A. Edison Guest of Honor — Merchandising and Salesmanship Outstanding Topic — Inter-connection of Power Lines, Coal Consumption and Electrification of Farms on Program

CONVENING at Atlantic City, N. J., May 17-21, will gather eight thousand representatives of the electric light and power industry to discuss the affairs and problems involved in the business of serving an ever-increasing public, which in the last year had added a million and a half new customers and now represents over a million customer-owners.

The present N.E.L.A. convention, the forty-ninth, will accordingly consider all vital issues confronting the industry's immediate problems and future plans of growth. It is realized that its service to eighteen million patrons will not remain at that figure, but that with time, and at that a comparatively short while, a great expansion and development will have to be met and solved.

Among the problems to be discussed along these lines will be the development of power reserves through interconnecting power lines; the leasing of Muscle Shoals for power service; the engineering advances which have produced as much electricity from one and nine-tenths pounds of coal in 1926 as was produced in 1920 from three pounds of coal; the running of electric lines into farm sections and the development of improved urban service.

First General Session

With the exception of the afternoon session of the Accounting National Section on Tuesday, the day of this publication, all the business sessions will be held on the Million Dollar Pier.

The Hon. Edward L. Bader, Mayor of Atlantic City, will open the first general session on Tuesday morning with a welcoming address. This will be followed by the presidential address of President Davidson. Treasurer W. A. Jones and Managing Director M. H. Aylesworth will both give their reports for the year. These will be followed by a report from M. S. Sloan, chairman of the Public Relations National Section. Owen D. Young is scheduled next for an address. R. H. Ballard will have as the subject for an address, "What Is the Function of the N.E.L.A.?"

Public Policy Night

The principal speaker Public Policy Night will be Owen D. Young of the General Electric Co. Martin J. Insull, chairman of the Public Policy Committee, will present the report of that committee, and President Davidson, who is also chairman of the Charles A. Coffin Prize Committee, will give an address and present the award to the company winning the Charles A. Coffin Medal. Elisabeth Rethberg, soprano of the Metropolitan Opera, will sing upon the conclusion of Mr. Young's speech. Vincent Lopez and his orchestra will play before the formal opening of the meeting and also during the meeting between some of the addresses. The entire program will be broadcast over the radio.

Thomas A. Edison To Be Present

Thomas A. Edison is expected as the principal guest this year. His appearance at Atlantic City will be an event of wide attraction, as he has found it necessary for many years to refuse practically all invitations to conventions and public functions.

A long list of other notables, both in the industry, and in National and professional life, will be on the program. Merchandising and salesmanship will be the outstanding topic at this year's convention, with public relations and interconnection also receiving considerable space on the program. Among some of the invited speakers who will be on the convention program, are Edward L. Bader, Mayor of Atlantic City; Honorable James J. Davis, secretary of the United States Department of Labor; B. C. Forbes, editor of "Forbes Magazine;" A. G. Paterson, president of the National Association of Railroad and Utilities Commissions; Bruce Barton, advertising man and writer; Mrs. Mary Kink Sherman, president of the General Federation of Women's Clubs; S. H. Thompson, president of the American Farm Bureau Federation; Robert H. Montgomery, of Lybrand, Ross Brothers & Montgomery; S. W. Parr, University of Illinois and Dr. T. J. Grayson, University of Pennsylvania.

Case School Gets a Million; To Erect M. E. Building

Progress of the campaign to raise funds for a new mechanical engineering building and for additional endowment for the Case School of Applied Science, Cleveland, Ohio, has been announced by Dr. Charles S. Howe, president of the school.

Answering a call to raise $300,000, the alumni quota and the condition of a gift of $500,000, more than 60 per cent of former students have subscribed a total of $404,000 to date; and donations from the remaining students continue to pour in.

The original gift initiating the present development of the school was for $500,000, made by Charles W. Bingham. When this was announced, his son, Charles W. Bingham II, duplicated the sum. In this way the first million was obtained. With half of the total sum the mechanical building will be started immediately, and other projects of progress will soon be undertaken.

Government Calls Meeting on Valve Standardization

Following proposals for the reduction of the number of sizes of wrought iron and steel pipe, valves and fittings, submitted by the Manufacturers Standardization Society of the Valve and Fittings Industry, through its subcommittee on the simplification of product, the Department of Commerce will convene a general conference of technical experts, manufacturers, distributors and consumers, to consider means of standardization, the morning of May 28 at the Commerce Building, Washington, D. C.

For some time there has been in progress a survey of current practice, the ultimate purpose of which has been to eliminate superfluous sizes of the products arising from the pipe industry. This conference, at which the details involved and the development of a unanimous simplified practice recommendation will be studied, is the first step in trend long thought necessary.

The Department of Commerce is now issuing invitations to the meeting to all who are professionally interested in this matter. The problem, under the following headings, will be discussed:

I. Sizes of wrought-iron and wrought-steel pipe, valves and fittings in common use.
II. A recommended range of sizes for future practice.
III. The possibility of using this recommended schedule as a basis for further simplification.

Cheat Haven, Pa., Gets Water Power Development

The West Virginia Power & Transmission Co., a subsidiary of the West Penn interests of Pittsburgh, is nearing the completion of its initial power desvelopment on the Cheat River at Cheat Haven, Pa., on the state line between Pennsylvania and West Virginia, seventeen miles south of Uniontown.

The first of the four 16,000-kva. 133-r.p.m. waterwheel generator units that will make up the generating equipment of this power station, was tested and shipped from the East Pittsburgh Works of the Westinghouse Electric & Manufacturing Co. recently.

Coal Bill Modified, Ready; But Senators Homesick

Influenced largely by the speedy settlement of the general strike in Great Britain, the report from the Senate Committee on Education and Labor made by Senator Copeland to urge the passage of the coal regulatory legislation at this session of Congress, presents a distinct modification of its original provisions.

The clause providing for seizure and operation of the mines in an emergency has been dropped, the committee having decided the proposition extremely unwise. "It need not be pointed out," reads the report in reference to this attitude, "that grave constitutional questions are involved in this suggestion. It was the opinion of the majority of the committee that no good end would be served by forcing such issues in the absence of immediate necessity." How much this opinion was influenced by pressure from the coal associations, and other propaganda, must be a matter of conjecture.

Although this change curtails the originally proposed power of the President during a national emergency, the bill still provides that if necessary, he may revive the Federal Fuel Distributor act of Sept. 22, 1922. In substance the bill is built on the plan for a fact-finding committee of the Bureau of Mines of the Department of Commerce with the function of investigating conditions and arranging the amicable adjustment of all disagreements and difficulties between miners and operators.

With many members of both Houses impatient to leave Washington and the pressure for the bill lessened on account of the settlement of the British strike, it is doubtful that the bill will be pushed further this session.

District Heating Men Plan Broad Survey For Meeting

At the seventeenth annual convention of the National District Heating Association, meeting at the Niagara Hotel, Niagara Falls, N. Y., June 1-4, a number of papers and reports of special interest will be read and discussed by leading experts in the electrical and heating field.

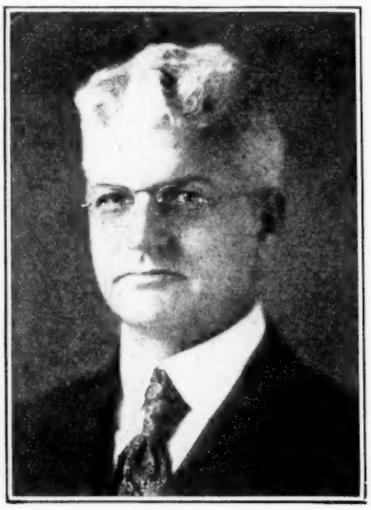
Outlining recent developments and gaging the future trend of central-station heating, a comprehensive study, illustrated by motion pictures, will be made through the presentation of six papers and addresses by James D. Hurd, New York Steam Corp.; J. H. Walker, Detroit Edison Co.; Charles H. Day, Cleveland Electric Illuminating Co.; T. E. Purcell, Allegheny County Steam Heating Co.; R. D. De Wolf, Rochester Gas & Electric Corp.; and L. E. Young, Union Electric Light & Power Co., St. Louis, Mo.

The report of the station operating committee will give the practical results of the use of powdered fuel and of fuel oil in heating, and also results obtained from electrical heated boilers in Canada. Committees will present reports on the following topics investigated: rates and regulations, operating statistics, distribution of hot water, heat utilization, and welding pipe.

R. Sanford Riley, Stoker Manufacturer, Dies

R. Sanford Riley, president of the Riley Stoker Corp., Worcester, Mass., and one of the most distinguished combustion engineers in the profession, died at his home in Worcester May 7.

Born in Hamilton, Ont., 1874, Mr. Riley was graduated with honors from the Worcester Polytechnic Institute in 1896. His early career included service as a draftsman with William Cramp & Sons, Philadelphia, and for a time he



R. Sanford Riley

was employed by the Baldwin Locomotive Works, Northern Pacific Railway, and as an instructor in mechanical engineering at the Worcester institute. In 1899 he enlisted as a marine engineer in the U. S. Navy and saw service in Asiatic waters. From 1906 to 1908 he was assistant to the chief engineer of the New York Shipbuilding Co., and was president of the American Ship Windlass Co., Providence, R. I., from 1906-11.

His most notable work was the development of the Taylor Stoker and the invention and development of the Riley stoker, and in 1912 he organized the Riley Stoker Co., which later purchased the Murphy Iron Works, Underfeed Stoker Co. of America, A. W. Cash Co. and United Machine & Manufacturing Co. There are also two foreign corporations which Mr. Riley headed, the Riley Engineering & Supply Co., Ltd., of Toronto, Canada, and the Riley Stoker Co., Ltd., of London, England.

Mr. Riley held a first-class engineer's license without tonnage limitation and during the war served as field officer of the Emergency Fleet Corporation, in which he organized the performance department and was in charge of all trial trips. He was formerly president of the Stoker Manufacturers' Association, and was a member of many engineering societies and civic organizations. He had been a member of the A.S.M.E. since 1906.

The passing away of Mr. Riley will be a matter of genuine sorrow to many people, not only in his home community but also throughout the entire engineering world. His death closes a career of much more than usual success.

Wide Activity In Pure Science Planned by Universities

Plans for a comprehensive national survey of research possibilities in pure science were made at a conference at the University of Pennsylvania in Philadelphia on Monday, May 3, which was attended by representative of universities, research institutions, foundations supporting research efforts, commercial laboratories and national scientific societies. The conference was held on the initiative of certain university heads and comprised men interested only in the physical sciences. Plans were made, however, for a larger meeting to be held next fall that will bring together those interested in every phase of rsearch.

The National Rsearch Endowmen, under the leadership of Secretary Herbert C. Hoover, Elihu Root and Charles E. Hughes, is engaged in a campaign to raise $20,000,000 to aid American universities in carrying on fundamental research during the next ten years.

One of the objects is to impress upon the public that applied science depends upon pure science and that pure science must be furthered if applied science is to grow and benefit the individual, the nation and industry.

President Josiah H. Penniman, of the University of Pennsylvania, presided at the conference and spoke of the increasing amount of attention paid in recent years to the development of research in universities and in other laboratories. He pointed out that discussions on the subject have arisen outside of university circles and that it is time for the universities interested in research to discuss how to promote its success.

On Again, Off Again— Muscle Shoals Bill

All indications are that the Muscle Shoals bill will not become a law at this session of Congress.

A very determined effort is being made, particularly by Southern Senators, to secure a vote on the measure recommended by the Joint Congressional Committee. This attitude is influenced largely by the desire to secure distribution of the power for public utility use. They are joined by others who feel that it is a reflection on Congress not to make some disposition of this matter which has hung fire so long.

The fact that the Joint Committee of the Senate and House should have recommended the offer of the Associated Southern Power Companies came as a surprise to most members of Congress. There had been strong farmer support for the offer of the American Cyanamide Co. Further support for that offer came from those who feel Muscle Shoals can be used to the greatest advantage for chemical manufacture.

It is very generally recognized, however, that the Joint Committee did a very good job of trading and has secured a bid from the power companies which it will be hard for any other bidder to meet, even were Congress to vote down the pending bill.

Papers of Interest For Power Men

Among important papers to be read at the annual convention and steel exposition of the Association of Iron and Steel Electrical, Mechanical, Combustion and Safety Engineers, to be held at Hotel Sherman, Chicago, June 7-10, will be a number of addresses and reports of special interest to men in the power field.

Refractories for steel mills will be discussed by M. S. Booze, Mellon University; W. J. Harper, Donner Steel Co.; M. J. Conway, Wheeling Steel Corp.; E. W. Trexler and W. H. Kelly, of the Bethlehem Steel Co.

A description of auxiliaries, boiler plant equipment, Nordberg reversing uniflow engine and operating control, together with results of tests of the Nordberg engine giving steam consumption per ton of steel rolled, will be discussed at length.

Explosion in an Enclosed Steam Engine

A somewhat unusual explosion recently in connection with a high speed inverted vertical compound steam engine of the enclosed crankcase type, with cylinders 16 in. and 29 in., stroke 7 in., speed 375 r.p.m., and employed forced lubrication, is described in the April issue of *Vulcan*.

Exact details of what happened are not available, as the engine attendant who was the only person in a position to throw light on the matter, sustained injuries, and could not be interviewed. It was evident, however, that an explosion of some severity had occurred within the crankcase.

The engine itself was not seriously affected, but the building in which it was housed caught fire and sustained coniderable damage. It is conjectured that heating up of one of the slippers in its guide was primarily responsible for the ignition of oil vapor within the crank case.

It was the general practice to leave one of the crank case doors slightly open for ventilation purposes. Oil vapor appears to have emerged from this partially open door and collected under the roof of the engine house in quantity. The primary explosion in the the crank case appears to have ignited the accumulated gaseous mixture under the roof and caused the ensuing damage to the latter.

Safety Rules for the Installation and Maintenance of Electric Utilization Equipment, consisting of Part 3 and the grounding rules of the fourth edition of the "National Electrical Safety Code," has just been issued by the United States Bureau of Standards, Washington, D. C. Copies of these rules can be had at fifteen cents each by writing to the Superintendent of Documents, Government Printing Office, Washington, D. C. Previous editions of the National Electrical Safety Code have been published in complete form. There has been a demand for smaller handbooks containing a single part of the code, and in response to this demand the fourth edition is being issued as a separate publication.

Flettner Demonstrates Pioneer Motor Ship To Trip Down New York Bay

Guests of German Inventor Are Passengers on Baden-Baden Which Shows Her Paces on Three-Hour Cruise

Anton Flettner's rotor ship "Baden-Baden," a recent arrival from Hamburg, Germany, was put through her paces last Thursday afternoon in New York Harbor for the benefit of a party made up of business men, engineers and journalists who enjoyed a three-hour cruise as guests of the German inventor. The weather was perfect, with enough of a breeze to demonstrate the capabilities of the funnel-like rotors which marked this ship as the strangest craft that ever visited these shores since Hendrick Hudson's "Half Moon" astonished the native Indians.

Propelled sometimes by the rotors alone and sometimes by the rotors in combination with the oil-engine-driven propeller, the "Baden-Baden" cruised around the harbor for three hours without mishap or difficulty of any sort before returning to her berth beneath the Whitehall Building at the lower tip of Manhattan Island. With rotors alone and a 15-knot breeze she made a speed variously estimated at from 5 to 7½ knots.

Wind Supplements Engine

The "Baden-Baden" (formerly called the "Buckau") was an auxiliary sailing ship until the masts were removed and rotors installed in the latter part of 1924. An article in the Dec. 30, 1924 issue of *Power* described this pioneer of all rotor ships and explained the principle of operation. She is 160 ft. long with a beam of 36 ft. and a draft of 11 ft. The two smooth vertical cylindrical rotors, 9 ft. in diameter and 50 ft. high, are driven at a maximum speed of 120 r.p.m. by individual 11-hp. motors mounted in the rotor pivots. Electric energy for these motors is taken from a 46-hp. Krupp Diesel-generator set, which supplies all the current used aboard the ship. A 250-hp. Diesel is installed for auxiliary propeller drive.

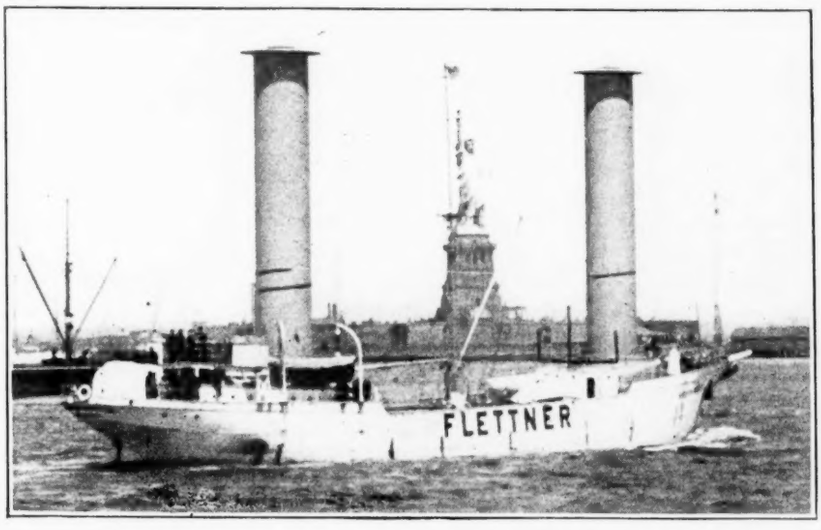
The spinning of the rotor so distorts the stream lines of the wind as to produce a partial vacuum on that side of the rotor which is moving parallel to the wind. This suction is kept forward by spinning the rotors clockwise when the wind is from the left and counter-clockwise when it comes from the right.

This side thrust is technically known as the Magnus effect, and has long been known to scientists. Sportsmen have also been familiar with the effect without, however, generally knowing its cause or its name. The Magnus effect explains the curving of base balls and golf balls, and the side drift of artillery projectiles. Anton Flettner was the first man to make useful application of this principle outside of the field of sport.

While sailing in the outer harbor the Baden-Baden made several tacks against the brisk breeze blowing up the harbor. She also ran part of the time with the wind on the beam. All this was without the propellor "Coming about" at the ends of tacks was facilitated by spinning the two rotors in opposite directions. On the return trip the wind was nearly astern, but the rotors were kept going and seemed to add materially to the speed imparted by the propeller.

Rotor Windmill Under Construction

Conversing with a member of the *Power* staff, Herr Flettner said the rotors were planned to supplement rather than replace the ordinary engine drive. Attention was called to launching on April 28 at Bremen of the 3,000-ton, 300-ft. long "Barbara," equipped with three rotors 55 ft. high and 12 ft. in diameter which replace about 1,200 hp. of engine capacity. She is expected to go into operation in July. A large power-generating rotor windmill is now being built for installation in Berlin.



Anton Flettner's rotor-ship Baden-Baden, driven by the action of the wind on her funnel-like rotors, plows past the Statue of Liberty

Southern Power Companies in Big Merger

Consolidation of fourteen Southern electric power companies with the Appalachian Electric Power Co., of West Virginia, was announced recently by C. S. Dawson, manager of the West Virginia Water & Electric Co., one of the concerns taken over in the merger.

The entire southern half of the State of West Virginia, the western half of Virginia and part of eastern Kentucky are now being served by these companies. The power plants of the systems are already connected by transmission lines.

The properties going into the consolidation are the following: West Virginia Water & Electric Co.; Appalachian Power & Light Co. (formerly the Virginian Power Co.); Kanawha Valley Power Co.; St. Albans Electric Power & Light Co.; Dunbar Light & Power Company; Interstate Power Co.; Consolidated Power & Light Co.; Kentucky and West Virginia Power Co. Inc.; Appalachian Power Co.; Roanoke Railway & Electric Co.; Lynchburg Traction & Light Co.; New River Development Co.; Central Virginia Power Co.; Boyd County Electric Co.

Detroit Edison Co. Plans Substation on Plant Site

In dismantling its old Farmer Street heating plant, the complete steam production of which aggregated less than the output of a single boiler in the new Beacon Street plant, the Detroit Edison Company plans to utilize the site for its electrical department. Service of the company's territory is therefore being carried from the four other heating plants of distribution.

The Farmer Street plant, located in a congested downtown district, has long had the problem of handling coal and ashes, and at the same time there has arisen acute need for a substation in the territory. For this purpose the old site will be used.

General Electric Co. will Quadruple Stock Shares

Stockholders of the General Electric Company, at the annual meeting in Schenectady, on May 11, approved the plan to change the present authorized common stock of 1,850,000 shares of the par value of $100 each into 7,400,000 shares of common stock without par value.

Rank of States Leading in Electricity Output, 1920-25

	Total Production—					
State	1920 Per Cent of U. S. Total	1921 Per Cent of U. S. Total	1922 Per Cent of U. S. Total	1923 Per Cent of U. S. Total	1924 Per Cent of U. S. Total	1925 Per Cent of U. S. Total
N. Y.	15.93	15.06	15.57	15.67	15.74	15.58
Pa.	9.90	9.86	9.50	9.77	9.68	9.91
Calif.	8.58	9.72	9.19	9.10	9.43	9.44
Ill.	6.98	7.28	7.31	7.31	7.91	8.00
Ohio	6.72	6.15	6.38	6.44	6.27	6.66
Mich.	4.81	4.64	4.57	4.64	4.57	4.60
Mass.	3.99	3.91	3.88	3.76	3.55	3.61
Wash.	2.75	2.87	2.70	2.60	2.55	2.46
W. Va.	2.59	2.69	2.84	3.07	2.99	2.49
Mont.	2.59	.....	.....	.....	.....	.....
Iowa	.....	2.41	.....	.....	.....	.....
N. J.	.....	.....	2.33	.....	.....	.....
Wis.	.....	.....	.....	2.28	2.42	2.34
Total	64.84	64.59	64.27	64.64	65.11	65.09

Personal Mention

Frank A. Tracy has been appointed assistant chief engineer of the Blackstone Valley Gas & Electric Co., Pawtucket, R. I.

Gilbert G. Rosenberg, mechanical engineer, with the General Electric Co., has been transferred from Schenectady to the Atlanta office.

Russell T. Waines has resigned from the faculty of the University of Toronto to join the testing department of the Cleveland Electric Illuminating Co.

Major Jones recently succeeded Major Paul S. Reinecke as District Engineer at Buffalo, N. Y. Major Reniecke was also a member of the Niagara Control Board.

Edward Rahm, formerly of the Erie City Iron Works, has accepted the position of sales engineer in the Pittsburgh district of the Reisert Automatic Water Purifying Company.

Charles Wohlers, formerly of Electric Bond & Share Co. and the Foundation Co., has recently joined the staff of Thomas E. Murray, Inc., consulting engineers, New York City.

Frank W. Peek, Jr., General Electric Co., Philadelphia, will be a recipient of the Levy Medal of the Franklin Institute, Philadelphia, at the medal meeting of the society, the afternoon of May 12.

M. T. Martin, formerly of the turbine engineering department of the Westinghouse Electric & Manufacturing Co., is now associated with Webster Tallmadge, specialist in steam engineering, 50 Church St., New York City.

Prof. W. Peterson of the Technical High School, Darmstadt, has been elected a director of the Allgemeine Elektricitäts-Gesellschaft of Berlin, in succession to the late Prof. Dr. Klingenberg.

C. W. Stokes, formerly a manager of the Sterling Engineering Co. at Montreal, and later the Canadian manager of the English Electric Co., Ltd., has now moved to Chicago to join the American Brown Boveri Electric Corporation.

C. J. Rodman, district manager of the Reisert Automatic Water Purifying Co., is now employed as general manager, secretary and treasurer, of the Buckeye Jack Manufacturing Co., of Alliance, Ohio. He has also recently been elected director and secretary of the Alliance Toy & Specialty Company.

Samuel Insull, president of the Commonwealth Edison Company of Chicago, has been suggested by Lord Rothermere, British publicist, as the logical selection for the post of Minister of Electrical Development. The post carries a seat in the Cabinet. The British Government has made no announcement of decision as yet.

Henry Eggelhof, P. O. Box 945, Dallas, Tex., was recently appointed exclusive representative for the eastern half of Texas by the Uehling Instrument Co., of Paterson, N. J.

Morgan B. Smith, chairman of the Detroit section of the A.S.M.E., has resigned as purchasing agent for the Detroit Edison Co., to become member-at-large of the Works Managers' Committee of the power division of the General Electric.

J. J. P. MacKenzie, formerly with the Broklyn Edison Company's engineering investigation bureau of the Mechanical Engineers' Department, has joined the engineering products division of the Walworth Company as assistant engineer. The division recently established headquarters in the Vanderbilt Building, New York City.

William H. Hartman, plant engineer, has resigned his post with the Kelly Island Lime & Transport Co., Marblehead, Ohio, to become chief engineer with the Bonnot Co., Canton, Ohio.

Business Notes

The General Electric Co., Schenectady, N. Y., has recently appointed R. M. Alvord manager of the San Francisco office. He succeeds E. O. Sheve, who was recently named manager of the industrial department of the company, with headquarters at Schenectady.

Stone & Webster, Inc., Boston, Mass., have been engaged by the Savannah Electric & Power Co. to proceed with the design and construction of an extension to its Riverside Power Station. This will consist of the installation of a 15,000-kw. turbo-generator, a 27,000-sq.ft. condenser and auxiliaries, and two boilers having a capacity of 1,370 hp. each with water-cooled walls.

The Crankless Engine Company of America, 29 Broadway, New York City, announces through Dr. E. H. Armstrong, of New York and Jacksonville, Florida, that contracts have been signed with a syndicate to manufacture in large quantities Crankless Diesel Engines for railway, marine and industrial uses.

The Riley Stoker Corp., Worcester, Mass., announces that O. C. Sheldon, formerly located at the Worcester office of the company as advertising manager, has been transferred to the Cincinnati office of the company as district manager; C. L. Smith, formerly district manager of the Cincinnati office, has been transferred to the Chicago office as district manager.

D. H. Skeen & Co., regulator department, Chicago, has been incorporated recently as the Mercon Regulator Co., for the manufacture of pressure-regulating valves of all types, in which a mercury column is used and marketed under the trade name of Mercon. The personnel of the company remains the same. The officers and directors are E. T. McCarthy, president; D. S. Walker, vice-president; G. L. Dickson, secretary; D. H. Skeen, treasurer.

The Timken Roller Bearing Service and Sales Co., eastern branch office, located at 1033 Cathedral St., Baltimore, Md., was closed May 1. The service requirements of Timken customers in this territory will be supplied through the Richmond, Pittsburgh and Philadelphia branches.

Frank Hodson, consulting engineer, 1015 Chestnut St., Philadelphia, formerly founder and president of the Philtric Furnace Construction Co., Philadelphia, advises that he has signed contracts with F. T. Kaelin, of Montreal, for the exclusive rights to the system, process and patents on the "Kaelin" Electric Steam Boiler for the United States and South America.

J. Barraja-Frauenfelder & Co., engineers, 8-10 Bridge St., New York, are announcing a consulting and advisory service on oil engines and their applications to the industries. The main branches of their service, classified, are as follows: Research and design; manufacturing advice; torsional vibrational investigations; marine and stationary power plants; appraisals, estimates, opinions and legal testimony.

The Power Specialty Co., 111 Broadway, New York City, announces that it has purchased the Aero Pulverizer Co. Through this acquisition the company is in the position to offer the purchasers of steam-plant equipment a contract with guarantees covering any boiler in combination with a water-cooled furnace of Foster design. Equipment furnished directly by this company will, therefore, now include Aero unit pulverizers, Foster superheaters, water-cooled furnaces, economizers and air heaters.

Trade Catalogs

Analysis of Missouri Coals—This is Technical Paper No. 366 and is published by the Bureau of Mines. It can be purchased from the Superintendent of Documents, Government Printing Office, Washington, D. C., for 10 cents.

Turbo Boiler-Feed Pumps—The Coppus Engineering Corp., Worcester, Mass. Bulletin No. 126 contains information on installing and operating the centrifugal turbo boiler feed pumps, type TB made by this company. Bulletin No. 125 covers types TB and TBM steam turbine and electric motor driven pumps. The bulletins are well illustrated.

Gaskets—Wilcox Manufacturing Co., East 77th St. and Anthony Ave., Chicago. Catalog No. 20 features several of the many varieties of gaskets made by this company and tabulates physical dimensions. Essentially, the gasget is a metal covering with an elastic filling, and, depending upon the use, the combination may be copper and rubber, copper and asbestos, steel or monel metal and asbestos, aluminum and asbestos, lead and asbestos or copper and leather.

Coming Conventions

American Boiler Manufacturers Association. D. W. Glanzer, 840 Rockefeller Bldg., Cleveland, Ohio. Annual convention at Homestead, Hot Springs, Va., June 1-2.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Regional meeting at Niagara Falls, May 26-28. Annual convention at White Sulphur Springs, W. Va., June 21-25.

American Order of Steam Engineers, W. S. Wetzler, 758 N. 44th St., Philadelphia, Pa. Annual convention at Moose Hall, Philadelphia, June 7-11.

American Society for Testing Materials. C. L. Warwick, 1315 Spruce St., Philadelphia, Pa. Convention at Haddon Hall, Atlantic City, N. J., June 21-25.

The American Society of Heating and Ventilating Engineers, A. V. Hutchinson, 29 West 39th St., New York City. Annual convention at Lexington, Ky., May 26-28.

American Society of Mechanical Engineers. Calvin W. Rice, 29 West 39th St., New York City. Spring convention at San Francisco, Calif., June 28-30.

American Water Works Association. W. M. Niesley, Assistant to Sec., 170 Broadway, New York City. Annual convention at Hotel Statler, Buffalo, N. Y., June 7-12.

Association of Iron & Steel Electrical Engineers. John F. Kelly, 1007 Empire Bldg., Pittsburgh, Pa. Exposition and convention at Hotel Sherman, Chicago, Ill., June 7-10.

Canadian Association of Stationary Engineers. G. A. Brown, secretary, Major St., Toronto, Ontario. Convention and exposition, London, Ontario, June 29.

Master Boiler Makers Association. Harry D. Vought, 26 Cortlandt St., New York City. Annual convention at the Statler Hotel, Buffalo, N. Y., May 25-26.

National Association of Stationary Engineers. F. W. Raven, 417 South Dearborn St., Chicago, Ill. Convention at Atlantic City, begins Sept. 14. Erroneously announced as Sept. 6-11. Annual conventions and exhibitions of state association are scheduled as follows: **Indiana State Association,** F. L. Clifford, 653 South Union St., Kokomo, Ind., Convention at Lafayette, Ind., May 10-15; **Kansas State Association.** H. S. Dukes, School for the Blind, Kansas City, Mo. Convention at Abilene, Kans., May 19-21; **Illinois State Association.** O. Jacobsen, 837, Lakeside, Place, Chicago. Convention at Decatur, Ill., June 2-4; **New England States Association,** Freeman L. Tyler, 32 Briggs St., Taunton, Mass. Convention at New Bedford, Mass., June 16-19; **Ohio State Association,** T. S. Garrett, 2622 E. Second St., Dayton, Ohio, Convention at Newark, Ohio, June 17-19; **Connecticut State Association,** George F. Klopfer, 30 East Pearl St., New Haven, Conn., Convention at Norwich, Conn., June 25-26; **Wisconsin State Association,** R. L. Scott, Eau Claire, Wis. Convention at Milwaukee, Aug. 3-6; **Minnesota State Association,** C. A. Nelson, 800 22nd Ave., N. E. Minneapolis, Minn. Convention at Rochester, Minn., Aug. 4-6; **New Jersey State Association,** S. G. Dalrymple, 111 Hutton St., Jersey City, Convention at Atlantic City, Sept. 12; **Pennsylvania State Association,** Frank J. McCarron, 3647 North 11th St., Philadelphia, Convention at Philadelphia, Sept. 12-13.

National Board of Boiler and Pressure Vessel Inspectors. Annual meeting at Chicago, May 24-26.

National District Heating Association. D. L. Gaskill, Greenville, Ohio. Convention at Niagara Hotel, Niagara Falls, June 1-4.

National Electric Light Association. A. Jackson Marshall, 29 West 39th St., New York City. Forty-ninth convention and manufacturers' exhibition, Atlantic City, May 17-21.

Universal Craftsmen Council of Engineers. 24th convention and exhibit Milwaukee Auditorium, Milwaukee, Aug. 3-7.

Plastic Furnace Lining—The S. Obermayer Co., Chicago. An attractive 28-page catalog devoted to "Ramtite," its various uses, methods of application and comparison with other materials.

Meters—Republic Flow Meters Co., Chicago. "Steam Economy" is the title of an instructive bulletin recently issued. A preliminary section on Steam Economy deals with operation and management, instruments required for high-grade boiler operation, the importance of records, measuring steam distribution and power plant cost accounting. This is followed by illustrations of typical installations in many industries and a final division devoted to the theory and construction of steam-flow meters, including typical arrangements of boiler panels.

Fuel Prices

COAL

The following table shows the trend of the spot steam market in various coals, f.o.b. mines; mine run except Pittsburgh gas slack:

Bituminous Net Tons	Market Quoting	May 10 1926
Pool 1	New York	$2.50@$2.75
Smokeless	Boston	1.88
Clearfield	Boston	1.75@ 2.10
Somerset	Boston	1.85@ 2.15
Kanawha	Columbus	1.40@ 1.75
Hocking	Columbus	1.40@ 1.75
Pittsburgh	Pittsburgh	1.75@ 1.90
Pittsburgh gas slack	Pittsburgh	1.50@ 1.60
Franklin, Ill.	Chicago	2.35@ 2.50
Central, Ill.	Chicago	2.00@ 2.15
Ind. 4th Vein	Chicago	2.10@ 2.25
West Ky.	Louisville	1.10@ 1.40
S. E. Ky.	Louisville	1.35@ 1.65
Big Seam	Birmingham	1.75@ 2.25
Anthracite Gross Tons		
Buckwheat No. 1.	New York	1.85@ 3.50
Buckwheat No. 1.	Philadelphia	2.00@ 2.75
Birdseye	New York	2.00

FUEL OIL

New York—May 13, light oil, tank-car lots; 28@34 deg. Baumé, 5¾c. per gal.; 36@40 deg., 6½c. per gal. f.o.b. Bayonne, N. J.

St. Louis—May 4, tank-car lots, f.o.b. St. Louis; 24@26 deg., $2.15 per bbl.; 26@28 deg., $2.20 per bbl.; 28@30 deg., $2.25 per bbl.; 30@32 deg., $2.30 per bbl.; 32@36 deg., gas oil, 6c. per gal.; 38@40 deg.., 6.7c. per gal.

Pittsburgh—May 11, f.o.b. local refinery; 30@34 deg., fuel oil, 6c. per gal.; 36@40 deg., fuel oil, 6⅝c. per gal.

Dallas—May 9, f.o.b. local refinery 26@30 deg., $1.60 per bbl.

Philadelphia—May 7, 27@30 deg., $2.31@$2.37 per bbl.; 18@22 deg., $1.93@$1.99; 13@19 deg., $1.68@$1.74 per bbl.

Cincinnati — May 11, tank-car lots f.o.b. local refinery, 24@26 deg. Baumé, 6½c. per gal.; 26@30 deg., 6⅝c. per gal.; 30@32 deg., 7c. per gal.

Chicago—May 10, tank-car lots f.o.b. Oklahoma, freight to Chicago, 92c. per bbl.; 24@26 deg., $1.40@$1.45 per bbl.; 26@30 deg., $1.45@$1.50; 30@32 deg., $1.60@$1.65.

Boston—May 10, tank-car lots, f.o.b. 12@14 deg. Baumé, 4.45c. per gal.; 28@32 deg., 5¾c. per gal.

New Plant Construction

Ariz., Chandler—City, F. V. Howey, Clk., will receive bids until May 27 for the construction of a waterworks system including well, 275 ft. deep, two 750 g.p.m. centrifugal pumps, 175 ft. head, direct connected to electric motors, 100,000 gal. tank on 110 ft. tower, etc. Weiland Engineering Co., Thatcher Bldg., Pueblo, Colo., is engineer.

Calif., Laguna Beach — Laguna Beach County Water Dist., P. O. Box 647, had plans prepared for the construction of two reservoirs and pumping equipment including two 900 g.p.m. deep well turbine pumps, four 900 g.p.m. and one 300 g.p.m. horizontal centrifugal pumps. J. P. Lippincott, Central Bldg., Los Angeles, is consulting engineer. E. A. Rowe, resident engineer.

Calif., Napa—Chamber of Commerce and F. B. Cox, is having preliminary plans prepared for the construction of a pre-cooling plant. Estimated cost $150,000.

Calif., Santa Monica — City is having plans prepared for the construction of a pumping station for outfall sewer including two 4,000 g.p.m. motor driven centrifugal pumps, etc. Estimated cost $35,000. H. B. Carter, is engineer.

Calif., Vallejo—Sacramento River Fishermen's Union, Pittsburgh, plans the construction of a packing and icing plant, here. Estimated cost $50,000.

Colo., Denver—P. A. Zang, 320 American National Bank Bldg., will soon award contract for the construction of an 11 story hotel at 454 South Marion St. Estimated cost $1,100,000. Hooper & Jarnsch, 879 North State St., Chicago, Ill., are architects.

Fla., Sarasota—West Coast Brick Corp., plans the construction of a plant, 50,000 brick daily capacity. Estimated cost $100,000. Owner is in the market for two 150 hp. tubular boilers, etc.

Ga., Savannah — Savannah Electric & Power Co., 27 West Bay St., awarded contract for the design and construction of extensions to Riverside power station, including 15,000 kw. turbo generator, 27,000 sq.ft. condenser and auxiliaries, two 1,370 hp. boilers, etc., to Stone & Webster Inc., 147 Milk St., Boston, Mass.

Ill., Chicago—The Astor Building Co., 116 North La Salle St., awarded contract for the construction of a 16 story apartment on Astor Ave. to Lundoff-Bicknell Co., B. F. Keith Bldg. Estimated cost $2,000,000. Steam heating system will be installed.

Ill., Chicago—Callner Construction Co., 177 North State St., will soon award contract for the construction of a 19 story apartment at Oakdale and Commonwealth Aves. Estimated cost $3,250,000. R. Gregori, 155 North Clark St., is architect. Owner is general contractor.

Ill., Chicago—Commonwealth Edison Co., 72 Adams St., will soon award contract for the construction of the second section of its power plant including boiler room, turbine room, transformer house, intake crib house, third section of switch house, etc., at 3500 South Crawford Ave. Estimated cost $2,000,000. Graham, Anderson, Probst & White, 80 East Jackson Blvd., are architects.

Ill., Chicago—Ridge General Hospital, Inc., 6542-6614 Ridge Blvd., is having plans prepared for the construction of a 6 story hospital including separate power house, etc. Estimated cost $1,500,000. P. T. Johnstone, 190 North State St., is architect.

Ill., Chicago—Sanitary Dist., H. E. Wallace, Clk., will receive bids until May 20 for 11 oil circuit breakers, electrically operated, etc., for Fullerton Avenue pumping station.

Ill., East St. Louis—Broadway Central Hotel Co., 700 East Broadway, awarded contract for the construction of a 7 story hotel at Broadway and Fifth St., to Hollenbery & Co., 118 La Salle St., Chicago. Estimated cost $1,100,000.

Ill., Evanston—The Toy Tinkers, Inc., is in the market for a 750 a. 6 v. or 375 a. 12 v. motor generator set for electroplating plant.

Ind., Evansville—H. F. Koch, Outer Lincoln St., had plans prepared for the construction of a 12 story office building. Estimated cost $600,000. Private plans.

Ind., Gary—Little Bros Co., is in the market for an ice machine, etc., for proposed new market at La Crosse, Wis.

Ind., Terre Haute — Treasury Dept., Washington, D. C., will receive bids until June 9 for a boiler for post office here.

Ia., Indianola — City is having preliminary plans prepared for the construction of either wells or storage reservoir including pumping equipment, distribution system, etc., for water supply. Currie Engineering Co., Webster City, is engineer.

Ky., Paducah—Treasury Dept., Washington, D. C., will receive bids until June 7 for a boiler for post office here.

La., Alexandria—Arkansas Oak Flooring Co., E. F. Ensign, Memphis, Tenn., will build a new mill and power plant, including two 100 hp. boilers, pumps, 750 kw. generator, air compressors, 60,000 gal. tank, etc. Work will be done by day labor. Estimated cost $400,000.

La., New Orleans—Syndicate, c/o E. L. Aschaffenberg, Chn., awarded contract for the construction of a 12 story hotel at St. Charles Ave. and Josephine St., to J. V. & R. T. Burkes, Carondelet Bldg. Estimated cost $1,000,000.

Me., West Buxton—Cumberland County Power & Light Co., awarded contract for the construction of a power plant to Foundation Co., 120 Liberty St., New York, N. Y. Estimated cost $500,000.

Md., Hagerstown—Bd. of Water Commissioners, W. P. Lane, Pres., will soon award contract for the construction of a filtration plant and pumping station. Estimated cost $1,500,000. G. L. Bean, Abbott Bldg., Philadelphia, Pa., is engineer.

Mass., Boston—Bureau of Yards & Docks, Navy Dept., Washington, D. C., will soon receive bids for the construction of a pump house, motor driven centrifugal pumping units, motor control equipment, etc., at Navy Yard, here.

Mass., Boston—State Health Dept., State House, plans the construction of a hospital. Estimated cost $500,000. Architect not selected.

Mass., Brockton — Edison Electric Illumination Co. plans to double transmission facilities from Dupont Circle sub-station to lines of Montaup Electric Co. about 14 miles, also extensions to switch house, bus and oil circuit breaker rooms, condenser room and outdoor sub-station including the installation of a 7,500 kva. synchronous condenser, station service transformers, etc. Stone & Webster, Inc., 147 Milk St., Boston, Mass., are engineers.

Mich., Detroit—Bd. of Water Commissioners, G. H. Fenkell, Gen. Mgr., 176 East Jefferson Ave., is having preliminary plans prepared for the construction of a booster station. Ultimate pumping equipment will probably include two 12,000,000 gal. and two 6,000,000 gal. electrically driven centrifugal pumps, etc.

Miss., Purvis — City State Investment Co., 128 North Wells St., Chicago, Ill., has acquired water and light plants and awarded contract for improvements consisting of a power house to include a new 100 hp. oil engine, 12 ton ice plant and equipment for remote control of two 25 hp. motors direct connected to two 350 g.p.m. water pumps, 125,000 gal. tank on 100 ft. tower, distribution mains, transmission line, etc., here, to A. Garrett, Canal Commerce Bldg., New Orleans, La. Estimated cost $125,000.

N. Y., Kings Park—State Hospital Commission, Capitol, Albany, will receive bids until May 26 for an air compressor and two electric generators for Kings Park State Hospital, here.

N. Y., New York—A. Adelson, 690 6th Ave., awarded contract for the construction of a 21 story office and loft building on Park Ave., to Shroeder & Koppel, 347 Madison Ave. Estimated cost $4,000,000.

N. Y., New York—Bing & Bing, 119 West 40th St., will build a 15 story apartment on East 57th St., by day labor. Estimated cost $1,000,000. E. Roth, 110 West 40th St., is architect.

N. Y., New York—Peter Cooper Lexington Corp., c/o R. Candela, 200 West 72nd St., Archt., is having plans prepared for the construction of a 15 story hotel at Lexington Ave. and 39th St. Estimated cost $900,000.

N. Y., New York—Five Hundred Seventy One Park Avenue Corp., c/o G. F. Pelham, 200 West 72nd St., Archt., will build a 15 story apartment at Park Ave. and 63rd St. by day labor. Estimated cost $1,000,000.

N. Y., New York—W. Warren, 16 East 47th St., will soon receive bids for the construction of an apartment on 5th Ave. Estimated cost $1,000,000. Warren & Wetmore, 16 East 47th St., are architects. H. Balcom, 16 East 47th St., is engineer.

N. Y., Ray Brook—State Hospital Commission, Albany, is having plans prepared for the construction of a power plant for New York State Hospital, here. Estimated cost $200,000.

N. Y., Rochester — Greater Rochester Properties, 207 Lafayette Bldg., Buffalo, is receiving bids for the construction of a 2 story theatre, office and stores building on South Clinton St., here. Estimated cost $1,125,000. L. Lempert, 1154 Mercantile Bldg., is architect.

O., Cleveland — G. Stewart, Pittsburgh Bldg., St. Paul, Minn., Archt., will receive bids about June 1 for the construction of a 10 story apartment and lodge building at East 40th and Windsor Sts., for Fraternal Order of Eagles, T. J. Long, Pres., Aeria No. 135. Estimated cost $1,400,000.

Pa., Abington—Bd. of Education, c/o C. M. Doll, Glenside, will receive bids until May 24 for the construction of a power plant for high school here. Heacock & Hokenson, 1211 Chestnut St., Philadelphia, are architects.

Pa., Philadelphia—Manayunk Plush Co., 109 Levering St., will soon award contract for the construction of a dye house, finishing house and power plant at Ambria and Lemonte Sts. W. E. S. Dyer, Land Title Bldg., is engineer.

Pa., Pittsburgh — Pittsburgh Insurance Exchange, c/o W. M. Reed, Union National Bank Bldg., awarded contract for the construction of a 21 story office building on Fourth Ave., to Boyle Robertson Co., 815 15th St., N. W., Washington, D. C. Estimated cost $1,250,000.

S. D., Sioux Falls—Bd. of Commissioners will receive bids until May 25 for a 400 hp. oil engine with 270 kw. generator and 3,200 g.p.m. duplex acting power pump against 190 ft. head. Maury & Gordon, 1445 Monadnock Block, Chicago, Ill., are engineers.

Tex., Brownsville—Bd. of Education, is receiving bids for the construction of a power plant. Estimated cost $40,000. H. P. Smith, National Bank of Commerce Bldg., San Antonio, is engineer.

Tex., Childress—City plans an election soon to vote $35,000 bonds for extensions and improvements to waterworks including pumping stations, mains, etc.

Tex., Goliad—Texas Central Power Co., N. B. Weiswanger, Mgr., Frost Bldg., San Antonio, is having plans prepared for the construction of a 40 ton ice plant and an electric power plant, here. Estimated cost $50,000 and $40,000 respectively. Private plans. Machinery for both plants will be required.

Tex., Shelbyville—Nacogdoches Ice Cream Co., plans the construction of a cold storage plant. Estimated cost $35,000. Private plans.

Wis., Edgerton—City, C. A. Hoen, Clk., will receive bids until May 25 for drilling new well, construction of a pumping station and installation of one 500 g.p.m. motor or engine driven centrifugal pump, etc. W. G. Kirchoffer, 22 North Carroll St., Madison, is engineer.

Wis., Milwaukee—Chain Investment Co., c/o Rosman & Wierdsma, 490 Broadway, Archts, will soon receive bids for the construction of an 8 story apartment on Grand Ave. Estimated cost $600,000.

Wis., Milwaukee—Dept. of Public Works, R. Stoelting, Comr., will receive bids until May 20 for foundations for addition to boiler room, etc., at North Point pumping station. Estimated total cost $400,000.

Que., Verdun—City, J. Wishart, Secy., will soon receive bids for improvements to electric light plant. Estimated cost $26,000.